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FACTORS IN THE SELECTION OF LOCOMOTIVES IN RELATION TO THE ECONOMICS OF RAILWAY OPERATION

By O. S. Beyer, Jr.

ABSTRACT OF PAPER

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- The service
- The nature of the business
- Topography, train speeds, and train resistance.

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FACTORS IN THE SELECTION OF LOCOMOTIVES IN RELATION TO THE ECONOMICS OF RAILWAY OPERATION

By O. S. Beyer, Jr., Chicago, Ill.[1]

Non-Member

The problem of locomotive design is comparatively simple when it is clearly known what is desired. The possibility of effecting operating results by the introduction of improved locomotives alone, or by their use in connection with such changes as grade revision, is not as fully appreciated as it ought to be. To make an intelligent selection of motive power for a railroad, it is necessary to study the effect which various types and sizes of locomotives will have on operating expenses and fixed charges. Statistics published by the Interstate Commerce Commission covering the entire railroad field of the United States show that 55 per cent of the operating expenses are affected more or less directly by the motive power.

2 The wide range of motive power now available will have to be considered in future double track and relocation work, grade reduction, elimination of rise and fall, and curvature and distance, in order to effect the greatest economy possible for the capital expended. It should no longer be necessary when relocating a division to increase its capacity and reduce operating expenses, to go to extremely heavy capital expenditures to reduce grades to the minimum of 0.2 or 0.3 per cent., as it is usually much cheaper to provide locomotives of greater power.

3 The main steps in the careful selection of motive power may be divided into the consideration and study of:

a the service

b the nature of the business

[1] Care of 2d Vice-Pres. Rock Island Lines, La Salle St. Sta.

The American Society of Mechanical Engineers, 29 West 39th Street New York. All papers are subject to revision.

c the topography of the road, train speed and train resistances
d the types and sizes of locomotives available
e improvements to the permanent plant
f effect of various types and sizes of locomotives on operating expenses
g final selection of most economical type and size of locomotive

THE SERVICE

4 Motive power is used in three general classes of service, namely, passenger, freight, and switching.

5 Passenger service may be further subdivided into the following classes: High speed or through service, and suburban or local service.

6 Freight service may be subdivided into slow or drag service, fast freight service, and pusher service.

7 Consideration of the service will determine whether the engines are to be built suitable for just one class or for a combination of classes, such, for instance, as switch and helper service, slow and fast freight service, freight and helper service, or passenger and freight service. This goes hand in hand with the consideration of some of the other features, notably the amount of and the different kinds of business, the size and types of engines available, and the effect of the engines under consideration on operating expenses.

NATURE OF THE BUSINESS

8 The special nature of the business, while intimately related to the general service for which the engines are intended, deserves some particular consideration. Through passenger service, for instance, may vary between long distance high-speed hauls, and slow speed frequent stops hauls. Freight service, especially, may vary between hauling almost exclusively trains of low class bulk commodities, such as coal and iron ore, which generally move in well built high capacity steel cars at slow speeds, and hauling trains of a decidedly mixed nature, loads and empties, box, furniture, flat, automobile, gondola and stock cars, which are not of high carrying capacity and are seldom fully loaded, resulting in trains of great length when any large amounts of tonnage are hauled. Due to the generally inferior

construction of cars which compose a mixed train, the weakness of the draft rigging of the older cars, the limitations of the older types of air-brake equipment, and the tendency to long trains when heavy tonnage is handled on railroads where this class of business is general, the motive power problem assumes an aspect entirely different from that on roads where the business is of the nature first mentioned.

9 In switching service the demand may vary between straight yard work and hump work, and in pusher service it may vary between passenger pusher and freight pusher work.

TOPOGRAPHY, TRAIN SPEED AND TRAIN RESISTANCE

10 The consideration of the topography, train speeds, and train resistance of the territory in which the new engines are to operate will at this time only take recognition of hauling capacities and train lengths. The extensive research work recently conducted by many eminent investigators, railroads, and locomotives builders furnishes reliable information and data from which to determine very exactly what the hauling capacities of different types and sizes of locomotives are, and the speeds and resistances of the trains which can be hauled by them.

11 The hauling capacity at different speeds depends on the sustained tractive effort, which in turn depends upon the boiler capacity and the engine efficiency. Locomotives intended for high-speed service should have high sustained tractive efforts. Locomotives for heavy drag and switching service should have high tractive efforts at slow speeds, and may sacrifice high sustained tractive efforts at high speeds in order to attain this end. When selecting locomotives to meet certain conditions in regard to train weights, train speeds and opposing grades, it is necessary to analyze the situation from this standpoint.

12 *Passenger Engines.* Grades, speeds, and train resistances must be very thoroughly considered when new passenger engines are to be purchased. Passenger engines are intended for high-speed work. Modern operating conditions frequently present cases of heavy trains composed of all steel cars. The weight of passenger cars, due to the desirability of greater passenger carrying capacity and heavier construction, is rapidly increasing. Hence, the most essential quality to be provided in a passenger engine is the ability to maintain large sustained

tractive efforts at high speeds, as well as high starting efforts at low speeds. This ability of a passenger engine is secured by providing ample boiler capacity and good steam engine efficiency. The ability of a passenger engine to accelerate rapidly is another quality which must be considered, particularly if the engine is to be engaged in service in which stops are frequent. The power required for high rates of acceleration, especially on grades, with heavy passenger trains is enormous. Since the locomotive power available is limited principally by the steaming capacity of the boiler, the larger this steaming capacity the sooner will the engine be enabled to reach its maximum speed.

13 *Freight Engines.* Freight and pusher engines are engaged in a service which requires at the critical time very high tractive efforts at slow speeds. This depends to a large extent upon the weight placed on the drivers and the total weight of such engines must thus be so distributed that a relatively large weight falls on the drivers. With the cylinder and driver dimensions and the boiler pressure so arranged that the maximum tractive effort at the rim of the drivers equals 22 to 25 per cent of the weight on the drivers, a satisfactory combination for an engine for ordinary freight service is secured.

14 Modern conditions demand an increase in fast freight service and relatively large sustained tractive efforts at high speeds over heavy grades are becoming more necessary than ever. Locomotives for fast freight service only may afford to sacrifice some initial tractive effort for the sake of having recourse to a proportionately larger heating surface when great quantities of steam are necessary at high speeds. Pusher engines and road engines, on the other hand, intended exclusively for slow service may be permitted to have a large tractive effort capacity at slow speeds with a sacrifice in high sustained tractive efforts at high speeds.

15 The large majority of freight engines purchased are intended for a service which is a combination of fast freight and drag or slow freight service. Under these circumstances it is usually desirable to get as large a steaming capacity as possible consistent with a high tractive effort at low speeds. Incidentally, engines with good steaming capacities when operated with heavy tonnage trains are proportionately more econ-

omical in fuel and water consumption, as well as capable of maintaining a higher average speed over the entire division.

16 Consideration of the topography of the railroad and the hauling capacity of freight locomotives presents the problem of permissible train lengths. If the territory over which the locomotives are to operate is, to a large extent, a succession of many sags and pitches and contains several momentum grades, the continual running in and out of the slack between the cars is a serious matter and tends to limit the length of the train and the amount of tonnage per train which can be hauled. The nature of the business, whether it is ore, coal or pig iron moving in high capacity modern steel cars, or general merchandise moving in box cars, refrigerators, small capacity gondolas or other similar cars has a further bearing on this feature. The lower the average total car weights of trains the longer the trains when large amounts of tonnage per train are hauled. The introduction of high capacity friction draft gears, steel underframes, and improved air brakes are tending steadily to minimize the difficulty of operating very long trains. The perfection of the variable load brake may help further in this direction. Cases are on record where trains of 90 and even more loaded cars, each car weighing 70 tons, have been hauled with success. Trains of 125 empty cars are not unusual in daily operation. Most of these extremely long trains are running over territories whose grades are low, or which have no broken profile. Furthermore the nature of the lading hauled in these exceedingly long heavy trains, is such that, should any heavy shocks occur, the lading cannot be damaged. Cars filled with automobiles, furniture, or general merchandise or flat cars loaded with agricultural machinery are different and must be handled with greater care in shorter trains. Every district for which motive power is intended presents certain little peculiarities when advisable train lengths are considered. Experience in the operation of trains hauled by the older types of locomotives must assist in determining the final answer to this question. As a rule there are a great many circumstances, such as the capacity of the side tracks and other conditions of the permanent way, which are apt to limit the length of trains before the profile conditions establish limitations.

SIZES AND TYPES OF LOCOMOTIVES

17 *Passenger Locomotives.* The principal types of locomotives available for passenger service are the Atlantic (4-4-2) type and the Pacific (4-6-2) type. The American (4-4-0) type, the Ten-Wheel (4-6-0) type, and the Prairie (2-6-2) type have been employed to some extent, but except in unusual cases are not of special advantage. The latest development in passenger engines for severe mountain service is the Mountain (4-8-2) type. In some special cases, such as exceedingly heavy mountain service, the Mallet engine has been used.

18 The Atlantic (4-4-2) type locomotive is usually best adapted to a service in which the trains weigh about 300 to 350 tons behind the tender and the grades encountered are relatively light. Owing to its wheel arrangement it permits of a boiler of ample capacity in proportion to the cylinders. It has a short rigid wheel base and a short total wheel base. To get very high initial tractive efforts with locomotives of this type means exceptionally high axle loads, which are undesirable. Hence, when high initial and sustained tractive efforts are required for heavy trains weighing over 350 tons behind the tender and operating over heavy grades, the Pacific (4-6-2) type is usually found to be better. The wheel arrangement of this type permits a still larger boiler, greater total weight on drivers with lower average axle loads, and larger cylinders. In general, the full utilization of these features results in both higher initial and higher sustained tractive efforts, combined with better accelerative qualities.

19 For exceptionally heavy mountain service the Mountain type permits of still larger boiler capacities and greater total weights on drivers, and hence still higher tractive efforts. Several engines of this kind are in service on the Chesapeake and Ohio Railway hauling trains weighing 600 to 650 tons over grades of 70 ft. per mile at 25 to 26 miles per hour.

20 Mallet engines have been introduced to a limited extent in passenger service. The Central Pacific Railroad has 12 Mallets which are hauling passenger trains on grades 116 ft. per mile and 40 miles long. The Atchison, Topeka and Santa Fe Railway has two Mallet compound engines which were built to haul passenger trains. The service in both these cases is very severe and rather exceptional.

21 The Atlantic and Pacific type engines, under modern operating conditions, are, for high-speed and high-capacity passenger service, the most desirable types. Under certain special circumstances, long continuous opposing grades may justify compounding in connection with these engines. The introduction of the high temperature superheater and the sectional brick arch have helped materially to increase the capacity, fuel economy, and efficiency of passenger engines. The limitations of Atlantic and Pacific type passenger engines are principally controlled by the permissible wheel loads. When 60,000 to 63,000 lb. per pair of drivers is once reached, it is questionable, from many points of view, whether it is wise to go still higher. Hence, when greater tractive efforts are necessary than can be secured from an engine with 180,000 to 190,000 lb. on drivers it becomes a question of either reducing schedules, double heading, or introducing locomotives with an additional pair of drivers. Table 1 summarizes the principal dimensions of typical passenger locomotives.

22 *Freight Locomotives.* Recent developments have made available an exceptional field from which to select locomotives for freight service. It seems limited not so much by the extent to which it is possible to build freight engines as it is by the physical restrictions of the permanent way, the nature of the freight business hauled, length of trains, and topography of the road. These limitations are, of course, mostly very serious, and, as far as track gage is concerned, insurmountable, except perhaps in some special cases.

23 Many Moguls (2-6-0), Ten-Wheel (4-6-0) and Prairie (2-6-2) type locomotives are in freight service today. Their capacities, especially the Mogul and Ten-Wheel types, are hardly adequate for modern service conditions. The Prairie type, due to the possibility of equipping it with a liberal boiler and liberal grate area, has a few advantages over the others.

24 The type of locomotive which has been the standard on many of the American railroads in the past ten years is the Consolidation, or (2-8-0) type. It has been called upon to perform in services ranging from emergency passenger to slow heavy pusher and switching service. Engines of this type are being built for heavy and exacting freight service and their possibilities have not been exhausted. They utilize nearly the

TABLE 1 PRINCIPAL DIMENSIONS OF TYPICAL PASSENGER LOCOMOTIVES

Type	4-6-0	4-6-0	4-4-2	4-4-2	4-4-2	4-6-2	4-6-2	4-6-2	4-6-2	4-6-2	4-8-2	Mallet	Forney	Forney
Cylinders, diameter and stroke, in.	22½x26	22x26	21x26	17½x26	15x25x26	22x28	27x28	25x28	17½x29x28	27x28	29x28	25x38x28	20x24	20x24
Diameters of drivers, in.	69	69	79	73	73	77	79	73	73	80	62	63	61⅝	63
Diameters of boilers, in.	74	72	67⅜	68⅛	72	70	76⅜	77⅝	70	81¾	83¾	82	66	70
Boiler pressure, lb.	215	200	200	160	220	200	185	200	210	200	180	200	200	200
Firebox, length, in.	138	105	108⅛	102⅝	109⅝	108	114⅛	126	109⅝	120⅛	114⅛	120⅝	94	94
Firebox, width, in.	108	75	71¼	60⅝	63⅜	66	75¼	108¼	76¼	79¼	84¼	84	105	86
Tubes, number	398	400	347	24, 206	273	24, 173	36, 207	34, 252	26, 199	40, 241	40, 243	495, 424	447	365
Tubes, diameter, in.	2	2	2	5¼, 2	2¼	5⅜, 2	5½, 2¼	5⅜, 2	5½, 2¼	5½, 2¼	5½, 2¼	2, 2¼	1¾	2
Tubes, length, ft. and in.	15-3	14-11	16-10	18-0	14-6	20-0	22-0	20-0	21-0	22-0	19-0	20-6, 6-3	9-0	12-0
Wheel base, driving, ft. and in.	14-4	15-10	7-3	7-0	6-10	13-4	14-0	13-0	13-8	13-10	16-6	32-0*	12-6	15-0
Wheel base, engine, ft. and in.	25-6	26-10½	28-2	30-10	32-8	33-4	35-7	34-10	35-1	36-5	37-5	51-4	30-9	23-0
Wheel base, engine and tender, ft. and in.	55-1½	59-3	56-1	62-8	61-1	68-6¼	68-2½	67-4	66-11¾	71-5¾	70-6	85-1	.....	35-10
Weight, on drivers, lb.	171,00	158,000	105,500	116,000	112,125	141,500	172,500	179,500	163,500	197,800	239,000	320,100	120,860	137,500
Weight, of engine, lb.	217,000	209,000	200,000	202,000	231,675	221,100	269,000	284,000	268,800	317,000	330,000	384,800	201,700	229,000
Weight, tender, lb.	135,000	148,000	134,000	149,900	163,325	170,900	161,500	165,700	171,200	175,700	163,400	173,200	.....	...
Fuel	anthracite	soft coal	soft coal	soft coal	oil	oil	soft coal	anthracite	soft coal	soft coal	soft coal	oil	anthracite	peacoal
Heating surface, tubes	3158.5	3104	3041.3	2521	2318	2477	3800	3579	3233	4372	3795	6882†	1825.5	2275.4
Heating surface, superheater	....	...	...	479	....	580	...	821	619	988	845	....	....	...
Heating surface, firebox	228.5	199	185.5	194.5	190	181	220	239	210	253	337	235	156.3	159.3
Heating surface, total	3387	3303	3254.5	2715.5	2508*	2658	4048	3818	3443	4625	4132	7117	1981.8	2434.7
Grate area, sq. ft.	103.8	55	53.5	42.8	48	49.5	59.75	94.8	57.6	66.1	66.7	70	68.5	56
Tractive power, lb.	34,860	31,000	24,700	29,600	23,880	30,000	38,400	40,800	35,000	43,300	58,000	66,800	26,500	25,900

* In addition has 1147 sq. ft. reheating surface.

† Includes 1590 sq. ft. of feedwater heating surface.

total weight of the engine for adhesive purposes. A leading truck of two wheels only is provided permitting of a slightly extended boiler and taking from the drivers only weight enough to secure good guiding qualities. The steaming capacity, firebox size and grate area are necessarily limited, since the entire boiler and firebox must be carried over the drivers. The handicap imposed by the boiler limitations has not, until recently, been very serious.

25 Engines of the Consolidation type, having a maximum tractive power of 60,900 lb. are in service today. The diameter of their drivers is small, 54 in., and their total heating surface compared with the equivalent heating surface of a Mikado engine having the same tractive effort is but 70 per cent. The piston speeds of these large Consolidation engines, compared with the Mikado engine, are considerably higher.

26 The perfection of the high temperature superheater, the brick arch, and the Gaines combustion chamber opens up further opportunities for the Consolidation engine. The application of the superheater results in increased capacity which corresponds, roughly, to a 25 per cent larger boiler capacity than it was possible to provide in connection with saturated steam engines. The brick arches permit increased amounts of heat to be utilized from the fuel burned on restricted grate areas. It should be possible to build Consolidation engines with good steaming capacities and economical fuel requirements that can develop as high as 54,000 lb. maximum tractive effort.

27 An offshoot from the successful Consolidation freight engine is the 12-wheel or (4-8-0) type. This type has not been widely introduced. It has an undesirable ratio between total weight and adhesive weight. The increase in the length of boiler made possible by the four-wheel truck in place of the two-wheel truck of the Consolidation engine nets but little in the direction of increased boiler capacity. The increase in the heating surface of the boiler is at the wrong end. To improve the steaming capacity of the Consolidation engine it is necessary to introduce modifications at the firebox end.

28 The introduction of such modifications has resulted in the Mikado (2-8-2) type engine. By placing a trailing truck underneath the firebox better boiler construction becomes possible; also a decided increase in effective heating surface, a

deeper throat sheet and wider water legs are secured. However, as large a proportion of the total weight of the engine is not utilized for adhesive purposes as with the Consolidation type. By moving the firebox behind the drivers, it also becomes possible to enlarge the boiler diameter, and to increase the relative diameter of drivers, thereby permitting of lower piston speeds. The general construction of the Mikado locomotive is such that it permits of very ample steaming capacity and thus of high sustained tractive efforts. The application of the superheater and brick arch has further increased its capacity in this direction. It is most admirably suited to haul slow maximum tonnage freight trains one day and fast freight trains the next, a condition frequently met in railroad operation.

29 The size of Mikado locomotives for most roads is principally limited by the allowable weights on drivers. It seems to be generally considered that an individual axle load of 60,000 lb. for the better conditions of roadbed, as they are met with today, is very nearly the largest permissible. If so the Mikado engine, as far as size is concerned, has very nearly reached its limit, and the demand for still larger engines will have to be met either by introducing another pair of drivers, making five pairs in all, or by resorting to the Mallet type. With their weight on drivers limited to about 60,000 lb. per pair, it is possible to build Mikados which have a maximum tractive effort of 60,800 lb. with very favorable sustaining qualities at high speeds. The utilization of the superheater and brick arch in connection with the well proportioned boiler and efficiently designed engine make it possible for this size of locomotive to be operated without requiring excessive amounts of fuel so that one fireman can handle all that is needed.

30 To get still larger capacities than are provided by the Consolidation and Mikado types, the Decapod (2-10-0) and the Santa Fé (2-10-2) types are available. The Decapod (2-10-0) type, like the Consolidation and 12-wheel types, has limitations as regards boiler capacity, in consequence of which it is practically adapted to slow service only. Its high proportion of weight on drivers, giving it a high ratio of adhesion is of advantage for this kind of service. The Santa Fé type permits of better boiler proportions than those of the Decapod type, just as the Mikado is better than the Consolidation. The additional pair of drivers

enables a tractive effort about 20 to 25 per cent greater than can be secured from the Mikado engine. Allowing 60,000 lb. per pair, the maximum tractive effort possible should be about 73,000 to 75,000 lb., barring cylinder limitations. Several engines of this type now in service deliver a maximum tractive effort of 71,000 lb. It is reported that they can be handled by one fireman without unduly taxing him.

31 Locomotives with five pairs of coupled wheels have an exceedingly long rigid wheel base. This would introduce many complications should they be placed on territories where track curvature is frequent or severe. Furthermore, the exceptionally heavy pressures on the main pins and the heavy reciprocating parts justify expectation of maintenance difficulties. The long wheel base and the large number of heavy wheel loads in rigid order may be proportionately harder on the track than is the case with large Mikado engines.

32 Another type of engine which deserves consideration for freight service is the Mountain (4-8-2) type, which is similar to the Mikado in all its characteristics. Where fast freight service is abundant and high speed is frequent the additional advantages in guiding qualities secured by the four-wheel leading truck and the slightly increased boiler capacity are important.

33 The Pacific type engine for exclusive fast freight service, where grades are not severe and where this kind of service is heavy, is a very desirable type. A large number of these engines have been built for this service and are giving an excellent account of themselves.

34 The Mallet type offers quite as wide a field to choose from as the Pacific, Consolidation, Mikado, and Santa Fé types combined. Mallet locomotives have been built on both the compound and the simple principle. The wheel arrangement permits of a great number of practical combinations. The application of the superheater and brick arch, feedwater heater and reheater, together with well proportioned boilers and the compound feature has made possible units of large size and of good drawbar pull characteristics at different speeds. At the same time Mallets are economical in fuel consumption. The arrangement of the drivers in two independent sets, and the division of

TABLE 2 DIMENSIONS AND CHARACTERISTICS OF TYPICAL FREIGHT LOCOMOTIVES

Class	4-6-2	2-8-0	2-8-0	4-8-0	2-8-2	2-8-2	2-8-2	2-10-2	2-6-6-2	0-6-6-0	0-6-6-0	0-8-8-0	2-8-8-2	2-8-8-0	0-8-8-0
Cylinders, diameter and stroke, in.	26x26	25x30	24x32	24x30	28x30	28x30	29x28	30x32	22-35x32	22-35x32	24-37x32	26-41x28	27-27x28	28-42x32	26-41x26
Driving wheel, diameter, in.	69	57	54	56	63	63	56	60	56	56	55	51	56	63	56
Boiler, diameter, in.	74½	83¾	84	80	86	86⅛	83¾	88½	83⅞	86	84	90	87⅝	90	89⅞
Boiler, pressure, lb.	180	185	210	200	180	180	170	175	200	225	205	220	160	210	210
Fire box, length, in.	108⅛	126⅛	132	100⅛	108	108	114⅛	132	109	108⅛	117½	126⅛	144⅛	117¼	126⅛
Fire box, width, in.	75¼	114	40	64¼	84	84¼	84¼	96	97	96¼	96	114	96¼	96¼	114
Tubes, number	34, 248	38, 275	406	386	36, 238	43, 304	40, 238	30, 285	36, 243	36, 250	437	42, 270	45, 282	42, 332	38, 277
Tubes, diameter, in.	2, 5⅜	2, 5⅝	2¼	2¼	5½, 2¼	5¾, 2	5½, 2¼	6, 2¼	5½, 2¼	5½, 2¼	2¼	5½, 2¼	5½, 2¼	5½, 2¼	5½, 2¼
Tubes, length, ft. and in.	20	14-6	15-0	18-10	21-0	21-0	19-0	22-7½	24-0	21-0	21-0	24-0	24-10	24-0	24-0
Wheel base, driving, ft. and in.	12-6	17-0	15-7	16-0	17-0	17-0	16-6	20-9	10-0	10-2	29-8	14-9	15-6	43-3	15-0
Wheel base, engine, ft. and in.	33-7	26-1	24-4	27-1	35-2	35-2	34-10	39-8	48-10	31-2	29-8	40-2	57-5	52-6	40-8
Wheel base, engine and tender, ft. and in.	66-3½	63-1½	57-11¾	62-0	67-2½	67-3½	67-10½	74-4¼	79-3½	70-8¾	62-2¾	75-7¼	88-½	83-1	77-2¾
Weight in working order, total, lb.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
Weight in working order, drivers, lb.	172,500	228,500	225,200	213,200	243,200	236,500	242,000	301,800	337,300	352,000	350,100	457,000	435,500	420,000	468,500
Weight in working order, engine, lb.	268,000	254,000	250,300	261,100	318,850	312,000	323,000	378,700	405,000	352,000	350,100	457,000	483,000	450,000	468,500
Weight in working order, tender, lb.	144,400	166,700	142,100	160,900	161,150	159,700	168,000	183,300	158,000	174,000	129,900	168,800	186,400	150,000	184,200
Fuel	soft coal	anthracite	soft coal	soft coal	soft coal	soft coal	soft coal	soft coal	soft coal	oil	soft coal	soft coal	soft coal	soft coal	soft coal
Heating surface, sq. ft. tubes	3538.6	2841.6	3564	3922	4004	4592.8	3740	4841	4659.9	4160	5380	5245	5701	6120	5205.5
Heating surface, superheater	802	622.5	....	....	905	1085	832	970	984.8	858	....	1106	1257	1368	1002
Heating surface, water tubes	28.4	....	....	....	28	27.3	28	....	26.2	....	30	....	40	....	....
Heating surface, firebox	202	256.6	241	182	232	234	283	320	343.2	214	230	352.4	384	326	321.4
Heating surface, total, sq. ft.	3769	3098.2	3805	4041	4264	4854.1	4051	5161	5029.3	4374	5640	5597.4	6125	6446	5526.9
Grate area, sq. ft.	56.5	99.85	36.8	44.7	63	63.1	66.7	88	72.2	72.3	78.3	100	96.5	78.4	99.9
Maximum tractive power, lb.	39,000	51,750	60,928	52,400	57,100	57,100	60,800	71,500	72,800	81,900	76,700	105,500	99,200	100,000	105,000

the total engine weight over these two sets permits readily of meeting track and axle load limitations. Hence these engines offer a large field from which to make selection when the restrictions of the permanent plant are such that they cannot be overcome except by heavy expenditures.

35 Mallet engines can be built to deliver a maximum tractive effort of 140,000 lb. This would mean engines with ten pairs of drivers, each having an average load of about 60,000 lb. As long as 60,000 lb. remains the maximum average practical wheel load, while track curvature remains a consideration, and the gage of the track remains at 4 ft. $8\frac{1}{2}$ in., thereby limiting the height of the center of gravity of engines, it is questionable whether an engine much larger than this can be built. It is not a size which has been reached today, although there are Mallet engines in service which have ten pairs of drivers.

36 A large number of Mallet locomotives are in road and pusher service whose tractive effort working compound range from 73,000 lb. to 105,500 lb. They are meeting with success from the fuel, operating, and maintenance standpoints. The largest number of drivers under the engines referred to is eight pairs, the average weight per pair under the largest one being 58,560 lb. Hence, 105,500 lb. tractive effort is not far from the maximum possible with eight pairs of drivers allowing 60,000 lb. per pair.

37 Table 2 summarizes the principal dimensions and characteristics of typical freight locomotives.

38 *Switch Locomotives.* The types of switching locomotives available range from the six-wheel coupled to the ten-wheel coupled. Recently a Mallet compound engine has been placed in hump yard service by the St. Louis, Iron Mountain and Southern Railroad in order that long trains may be handled without breaking them up. Switching locomotives of five pairs of drivers have a rather long rigid wheel base, perhaps too long for the average yard conditions as they exist on many roads today. Locomotives with four pairs of wheels have a more suitable wheel base, and are capable of delivering comparatively high tractive powers. Locomotives of three pairs of drivers are the most universal in service today. Table 3 summarizes the general dimensions of typical switch engines.

THE PERMANENT PLANT AND ITS RELATION TO MOTIVE POWER SELECTION

39 The permanent plant of a railway as related to the motive power is the track, bridges, passing sidings, terminal yards, engine terminals, including the roundhouses, turntables, coaling stations, watering cranes, ash plant and sanding facilities, and the locomotive repair shops. It has been shown what a wide range of motive power is available from which selections may be made for any class of service. In order that the possibilities of this large field may be fully realized it becomes necessary to study carefully the various changes in the permanent plant to be considered in connection with the introduction of different types and sizes of engines. Such a study will oftentimes show that improvements made to the permanent plant at limited costs will permit of utilizing motive power which will effect a considerable saving in operating expenses, thereby fully justifying the expenditure.

40 *Track.* The improvement of the track with a view of making possible higher train speeds and heavier axle loads is a complicated problem. Track improvements such as laying heavier rails, respacing of ties, increasing the carrying power of the sub-grade by drainage, deepening and improving the nature of the ballast are continually under way. The tendency is to strengthen and improve track as rapidly as earnings will permit, so that full advantage may be taken of all that is offered in the way of possibilities to increase engine capacities and train tonnage. The state of good track today, laid with 90 and 100 pound rail, is such that it readily allows 60,000 lb. axle loads, which is about as high on the average as it is advisable to go. While the track may be such that, should heavy wheel loads be imposed, it would result in increasing the maintenance costs, the tendency may nevertheless be to improve this lighter track in the near future by introducing ballast and relaying the lighter rail with heavier. In such cases, and they are very prevalent, the present track conditions should not be permitted to limit the motive power sizes too greatly because of the economical advantages of increased train loads.

41 Bridges not only restrict the individual axle load, but also the total weight. Moderate expenditures in the direction of bridge improvement, plus the regular program of bridge

TABLE 3 GENERAL DIMENSIONS OF TYPICAL SWITCHING LOCOMOTIVES

Class	0–6–0	0–6–0	0–8–0	0–8–0	0–8–0	0–10–0	Mallet
Service	ordinary	ordinary	ordinary	ordinary	ordinary	hump	hump
Cylinders, in.	21 x 28	21 x 26	21 x 28	27 x 30	23½ x 32	24 x 28	26 – 40 x 32
Driving wheel diameter, in.	57	52	51	57	57	52	55
Boiler, diameter, in.	67⅜	80	68	83½	80	80	84
Boiler, pressure, lb.	180	200	180	165	200	210	200
Firebox, length, in.	72⅛	83⅝	114	111⅛	120	108	126
Firebox, width, in.	65¼	66	42¼	75¼	75¼	73	96
Tubes, number	304	400	260	450	511	447	401
Tubes, diameter, in.	2	2	2¼	2	2	2	2¼
Tubes, length, ft. and in.	16–0	11–0	14–2	15–0	14–9	19–0	21–0
Wheel base, driving, ft. and in.	11–6	11–0	15–0	16–0	16–0	19–0	39–4
Wheel base, engine, ft. and in.	11–6	11–0	15–0	16–0	16–0	19–0	56–7
Wheel base, engine and tender, ft. and in.	42–6	34–9¼	47–0	49–2¾	48–9½	54–5½	85–2¼
Weight in working order, lb.	......	......	......	......	......	......	......
Weight in working order, on drivers, lb.	166,400	165,000	164,000	229,000	223,950	270,000	395,000
Weight in working order, of engine, lb.	166,400	165,000	164,000	229,000	223,950	270,000	435,000
Weight, in working order, of tender, lb.	104,300	135,000	91,000	134,000	140,050	149,600	155,000
Fuel, kind	soft coal	soft coal	soft coal	soft coal	soft coal	soft coal	soft coal
Heating surface, tubes, sq. ft.	2535	2286	2157	3514.7	3924	4422.4	4281
Heating surface, feedwater heater tubes, sq. ft.	......	......	......	......	......	......	1230
Heating surface, firebox, sq. ft.	144	169	163	194.6	179	203	252
Heating surface, total, sq. ft.	2679	2455	2320	3709.3	4103	4625.4	5763*
Grate area, sq. ft.	32.6	38.5	33.6	58.1	62.7	55	84
Maximum tractive power, lb.	33,140	37,500	37,040	53,800	52,700	55,362	94,575

* In addition has 890 sq. ft. of superheater heating surface.

renewals and improvements may permit of taking advantage of a type of locomotive which would effect economies in operating expenses that would more than pay for the unusual improvement expenses incurred.

42 *Yards and Passing Sidings.* Unless the yards and passing sidings are adequate to take care of longer trains the improvements expected by the introduction of larger locomotives will not materialize and the operating expenses will show but little decrease. The long trains are apt to be tied up and blockades will occur which are expensive as well as demoralizing.

43 *Locomotive Terminals.* Except for side and overhead clearances, the coal chutes, watering cranes, sanding and ash handling facilities, will have but little effect on the choice of new locomotives. Ash pit construction may need a little modification to permit of heavier wheel loads. The roundhouse may be inadequate to permit of housing the larger engines suitably. The turntable may be too short or too light to turn the heavier engines. Limited or inadequate locomotive terminals, in part or in whole, should not be permitted to stand in the way of introducing the most economical locomotive available. It is becoming more generally recognized that as motive power increases in size it becomes relatively more important and economical to provide means at locomotive terminals whereby the turning of power is expedited.

44 *Shop Facilities.* It would be manifestly impractical for a railway to undertake to rebuild its entire shops for the sake of handling a very large engine of the Mallet type. As a rule, however, especially on the larger roads, the present shopping facilities are such that a railway, from this point of view, should find but few limitations to the size of locomotives it may find desirable to operate. Then again the adopted program of shop facility renewals and modifications, continually necessary as business keeps growing and old parts of the shops depreciate, tends to take away from the seriousness of this consideration.

45 *Clearances.* Clearances through cuts, of water cranes, buildings, and other fixed structures along the right of way have some effect in determining the size of the engine to be selected, as well as the amount of the expenditures to be made for improvement.

THE RELATION OF OPERATING EXPENSES TO THE SELECTION OF MOTIVE POWER

46 The effect of the selection of locomotives for passenger and switching service on operating expenses does not play as important a role as it does in the selection of engines for freight service. The choice of passenger and switch engines is determined very largely by imposed conditions resulting from circumstances peculiar to the nature of these two kinds of service. Larger and heavier switch engines are usually made necessary by heavier trains handled in the yards. The demand for larger and improved passenger engines results from the necessity of maintaining high speeds with trains which are growing in weight due to the introduction of steel equipment and more cars. It should be observed, however, that the economy of the various types of engines available, from the fuel, lubricating, and maintenance points of view has an important bearing on the ultimate selection. That type and size of engine should be chosen which will operate with the least expense for fuel, water, and repairs, and which will keep as low as possible, consistent with the size of the engine in regard to the service requirements for the present and the future, the expeditures for improvement to the permanent plant.

47 In the selection of power for freight service the effect of the various types and sizes on the operating costs should go a long way towards determining the most economical engine to choose. The largest part of the gross revenue of railways results from freight transportation, and the greatest proportion of their operating expenses are consumed in conducting this transportation. The nature of freight service permits a much greater flexibility as far as choice of the motive power is concerned. A study should be made to determine which type and size will effect the greatest net saving in operating expenses after deducting all overhead and additional maintenance charges resulting from the improvements necessitated by the introduction of the engine. Only by such a study as this in conjunction with considering the service conditions and the tendency of future development can the ultimate selection be made with any degree of correctness.

48 The items of operating costs and overhead charges which should be considered in such a study are the following:

a Transportation expenses
- Fuel
- Water
- Lubricants for locomotives
- Other supplies for locomotives
- Enginehouse expenses
- Train supplies and expenses
- Enginemen's wages
- Trainmen's wages

b Maintenance of equipment
- Locomotive repairs
- Freight train car repairs

c Maintenance of way and structures
- Ballast
- Ties
- Rail
- Other track material
- Labor, roadway and track
- Bridges, trestles and culverts
- Buildings, fixtures and grounds

d Overhead charges
- Interest on locomotives and improvements to permanent plant
- Depreciation of locomotives and improvements to permanent plant
- Taxes and insurance on locomotives and improvements to permanent plant.

49 To attempt to discuss within the limits of this paper the effect of locomotive types and sizes on each one of the items of railway expenses concerned would be impossible. Wellington, Webb, and Henderson have formulated many of the principles involved and reference to the works of these men will reveal considerable valuable data and information on this subject. The statistics kept by the individual railways will furnish additional information necessary for the analysis. The application of logical reasoning and the use of recent data established by both laboratory and service tests will further aid in making correct comparisons. The suggestions regarding the various items of

expenses and their relation to motive power selection will therefore be general and seek rather to point out the inherent tendencies of these expenses and their importance.

50 *Fuel.* Fuel is the largest single item of locomotive operating expenses and therefore the most important. The fuel consumption may be divided into the following classes: (*a*) fuel used while actually working on the road; (*b*) fuel used while drifting and waiting; (*c*) fuel used at terminals for firing up. As locomotives grow larger their fuel consumption per unit increases, but not nearly in proportion to the increase in their size. It does not take very much more coal to fire a large locomotive than a small one. The fuel losses of a large locomotive due to radiation while waiting or drifting are but slightly larger than those of a smaller locomotive. The increase of fuel consumption of large saturated simple steam engines when working at their full capacity is more nearly in proportion to the increase in their size. The introduction of the superheater, feedwater heater and reheater, the increase in heating surface of the boiler, the brick arch, the utilization of compounding in large engines of the Mallet type, application of improved valve gear and compound air pumps, and more careful attention to the design of steam passages and steam engine efficiency have accomplished remarkable results in keeping the fuel consumption of large locomotives down so that their consumption per train-mile is increased but slightly over that of the recent types of smaller saturated steam locomotives.

51 Numerous tests and service records have revealed that large superheater Mikado locomotives which have been placed in service recently haul trains of 45 and 50 per cent greater tonnage with the same amount of coal that was formerly consumed by the Consolidation locomotives they replaced. Even the coal consumption of Mallet engines with grate areas up to 100 sq. ft. has not grown in any way proportionate to the increase in their hauling capacity. Modern engines when running at shortened cut-offs over those portions of the road other than the ruling grades exhibited a still greater economy than when working on the heaviest grades. Some service tests of recently built Mikado engines on the Delaware, Lackawanna and Western Railroad clearly demonstrated these facts. Their economy in fuel consumption as compared with that of the old Consolida-

tion type, both operating over heavy grades at full load, was 20 per cent. The economy effected over easy grades while running at shortened cut-offs was 39.3 per cent, almost twice as much. The average was 29.1 per cent.

52 The conclusions to be reached in regard to the fuel consumption of larger locomotives equipped with those fuel saving devices which have proved their merit is that it increases but slightly as their hauling capacities increase. It depends of course largely upon the size of locomotives in service as to what the actual increase will be on the train-mile basis over the consumption of the old engines, and this must be taken into consideration.

53 *Water.* The same things that are true of fuel consumption, are true of water consumption, only perhaps more so, especially with superheater engines. Water economy increases but slightly with the increase in locomotive sizes.

54 *Lubrication of Locomotives.* It is safe to conclude that the cost of locomotive lubrication increases more or less directly in proportion to the increase in the size of the locomotive as reflected by the wheel and cylinder arrangements.

55 *Other Supplies for Locomotives.* This item is affected slightly, if any, by an increase in locomotive capacities.

56 *Engine House Expenses.* As the size of engines increases the cost of enginehouse expenses per engine handled increases. However, as the size of engines increases the number handled at the terminals decreases. This decrease should approximately counterbalance the increase in cost per engine handled.

57 *Train Supplies and Expenses.* The item of train supplies and expenses per train-mile is affected but slightly by a decrease in train mileage.

58 *Wages of Enginemen.* History clearly shows that the wages of engineers and firemen per trip increase as the engines grow larger. Prompter movements made possible by more powerful locomotives reduce the cost of overtime.

59 *Wages of Trainmen.* The wages of trainmen do not increase per train-mile. If the business is handled a little more promptly through the decrease in the number of trains handled there will be a corresponding decrease in the wages paid for overtime of trainmen.

60 *Locomotive Repairs.* The cost of locomotive repairs does

not increase in direct proportion to the increase in hauling capacity. Under modern conditions it will perhaps be safe to assume that about 60 per cent of the total cost of locomotive repairs varies directly as the weights of locomotives increase.

61 *Freight Train Car Repairs.* The item of freight train car repairs per train-mile increases slightly more than in direct proportion to the increase in the number of cars hauled in the longer trains by the larger locomotives. Improvements in car construction, however, are rapidly counteracting this tendency.

62 *Maintenance of Track, Bridges and Buildings.* General observations seem to indicate that about 40 to 50 per cent of the cost of maintaining track per train-mile varies directly with the average increase of weight on drivers. The many improvements in track construction continually under way, the introduction of treated ties, heavier rails of improved steel and better drainage facilities all help to reduce to a minimum the effect of heavier power on the roadbed. The improvements to bridges, buildings and structures, such as roundhouses, turntables and shops, the lengthening of passing sidings and the extension of yard facilities have a further effect on the cost of maintenance of way and structures as related to the increase of motive power sizes.

63 *Overhead Charges.* The overhead charges include interest, depreciation, taxes, and insurance and should be carefully estimated.

FINAL DETERMINATION OF MOST ECONOMICAL LOCOMOTIVE TO ADOPT

64 Taking each one of the items into consideration, estimating the reduction in train mileage effected by each type, the gross savings in operating expenses effected based on the amounts of business on hand or in sight, and deducting therefrom all overhead charges arising from the additional improvements necessary to make the operation of the different types of locomotive under comparison practical, will reveal which particular locomotive is the most economical in size and type.

65 As far as a standard engine of any kind for an entire road is concerned the general conditions obtaining will have some bearing. A road may for instance be composed partly of divisions whose grades are moderate and partly of divisions whose grades are severe. If the variations are not great a compromise standard might be adopted. If on the other hand there

is a large difference, it may be wiser to seek to establish two or three standards and confine them to their particular territory with a view to getting the maximum efficiency from every portion of the property. Then again there are many shorter territories such as pusher grades and divisions through mountainous country, the motive power selection for which is a distinctly local problem. In every case, whether it is the broad problem of establishing standards for the entire system, or selecting an engine for a local territory, the problem might well be reduced to an economic study, comparing several available types and sizes, the extent of the improvements necessary to make their operation practical, and the net savings which it is estimated will be effected by their introduction.

SUMMARY

66 In designing new locomotives all of the conditions must first be analyzed and then the design made to suit them. The actual design of the engine finally chosen may be approached with confidence because of accumulated knowledge and experience. Due to the great possibilities of favorably effecting operating results by building locomotives which are exactly suited to their work, a study of the conditions becomes vitally important. To show what these conditions are has been the object of this paper. The fact that the most powerful locomotives of most approved design are also the most economical should be more generally appreciated. It is to be hoped that the future will see more advantage taken of the modern locomotive in accordance with its possibilities in relation to grade revision and its ability to reduce operating expenses to a minimum. The ultimate benefits which will result will certainly be justified to the fullest extent.

THE PROPERTIES OF SATURATED AND SUPERHEATED AMMONIA

By William E. Mosher

ABSTRACT OF PAPER

In this paper the various scattered experimental data on the properties of ammonia are collected and correlated. An attempt is made to reconcile these data by means of well-known thermodynamic laws and principles so that the results may be consistent with each other.

The paper treats of both the saturated and the superheated vapor; also of liquid ammonia. Formulae to express the various properties are proposed, some being of rational form with empirical constants, others being wholly empirical. The constants in the various formulae are determined by aid of the experimental data. The values resulting from the use of these formulae are compared with the experimental values and substantial agreement is shown. Discussions of the validity of the different formulae are given.

The paper contains no working tables of the properties of ammonia, but such tables have been prepared for both the saturated and the superheated vapor, and they will appear, together with a Mollier diagram for ammonia, in due time as a bulletin of the Engineering Experiment Station of the University of Illinois.

THE PROPERTIES OF SATURATED AND SUPERHEATED AMMONIA

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Junior Member of the Society

The vapor of anhydrous ammonia first became of interest in the field of mechanical engineering with the advent of Carré's absorption and Linde's compression refrigerating machines. With the development of the refrigeration industry this vapor has become more and more important and an accurate knowledge of its properties is highly desirable.

2 As in the case of steam and many other vapors, the first reliable experimental knowledge of the properties of ammonia was derived from the work of Regnault. He determined experimentally the relation between the pressure and the temperature of the saturated vapor and expressed it by means of empirical formulae. He also determined the relative volumes of the superheated vapor at different pressures along an isothermal for the temperature 8.1 deg. cent., the specific heat, the theoretical density and the experimental density of the gas. The determinations made by Regnault of the specific heat of the liquid and the latent heat of vaporization were lost in the reign of the Commune in 1871; twelve of the determinations of the latter magnitude, however, have been found.

3 The tables now used most extensively in refrigeration work and included in most handbooks on refrigeration are those of Wood, based upon Regnault's experiments. Peabody's tables, which appeared at about the same time, were based upon the work of Ledoux and upon the same form of equation of state, namely that derived by Zeuner and applied to ammonia by Ledoux. Since these tables were published the properties of ammonia have been made the subject of considerable experi-

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The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

mental work and various tables have been issued, although comparatively little is known of the subject even now when its importance in technical work is considered. Probably the most accurate table for the range covered is one recently calculated by Goodenough, although this was designed for class room purposes and is not based upon refined analysis.

4 The present investigation was undertaken with the object of collecting and correlating the various scattered experimental data on the subject of the properties of ammonia. An attempt has been made to reconcile these data by means of well-known thermodynamic laws and principles so that the results may be consistent with each other; and to express the various properties by means of formulae from which tables and charts in English units may be prepared. It is realized, however, that the experimental evidence is not as complete as could be desired and that more accurate experimental work may render necessary slight revision of the values of the constants in the equations here presented.

5 *Notation.* Throughout the investigation the absolute temperature of melting ice on the Fahrenheit scale has been taken as 491.64 deg., so that absolute zero is at 459.64 deg. fahr. The value of 777.64 ft. lb. has been used for the mechanical equivalent of heat. The notation used is as follows:

J = Joule's equivalent.
A = Reciprocal of same.
t = Temperature on the F. or the C. scale.
T = Absolute temperature.
p = Pressure.
v = Specific volume.
γ = Specific weight.
u = Intrinsic energy.
i = Heat content at constant pressure.
s = Entropy.
r = Latent heat of vaporization.
ρ = Internal latent heat.
ψ = External latent heat $= Ap(v'' - v')$.
c = Specific heat.
c_p = Specific heat at constant pressure.
x = Quality of vapor mixture.

Subscript $_k$ indicates critical data.
$'$ indicates properties of liquid.
$''$ indicates properties of saturated vapor.

6 *Relation between Pressure and Temperature of Saturated Vapor.* To express the relation between the pressure and temperature of saturated vapors scores of formulae have been proposed, some being of more or less rational form involving empirical constants, but the greater number being purely empirical.

a Biot's equation as used by Regnault has the form

$$\log p = a - b\,\alpha^n + c\,\beta^n$$

where $n = t - k$

The same formula was used in the computation of Peabody's tables.

b Wood used Rankine's formula in the form

$$\log p = a - \frac{b}{T}$$

with $a = 6.2495$ and $b = 2196$; limits, -20 deg. fahr. to 100 deg. fahr.

c Goodenough used the Bertrand formula in the form

$$\log p = \log k - n \log \frac{T}{T - b}$$

with $\log k = 5.87395$, $n = 50$, $b = 84.3$.

7 It does not seem to be possible to find constants which will make equations of the above forms apply over more than a limited range of temperature, either with steam or with other vapors. An equation for steam has, however, recently been proposed by Marks[1], which is remarkable in that it gives pressures agreeing closely with experimental values throughout the range from 32 to 706.1 deg. fahr., the latter point being the critical temperature. This equation is based on the one used by van der Waals

$$\log \frac{p_k}{p} = a\left(\frac{T_k}{T} - 1\right) \qquad [1]$$

where p_k and T_k denote respectively the critical pressure and critical temperature. Instead of being a constant as used by van der Waals, a varies for different substances and for different temperatures with the same substance. Professor Marks found values of a at different temperatures for steam and then found an expression for its value

[1] Trans. Am.Soc.M.E., vol. 33, 1911, p. 361.

in terms of the temperature and the critical temperature. By substituting this expression for a and the proper values for p_k and T_k in equation [1], he arrived at the following relation between the pressure and temperature of saturated steam

$$\log p = a - \frac{b}{T} - cT + eT^2 \qquad [2]$$

where

$a = 10.515354$
$b = 4873.71$
$c = 0.00405096$
$e = 0.000001392964$

8 In the present investigation an attempt was made to use the above method to determine the pressure-temperature relation of ammonia. Several factors rendered the use of this method undesirable, among them being the sensitiveness of the method to changes in the critical data and the uncertainty of these data for ammonia, the lack of experimental data at temperatures near the critical point, and the large discrepancies existing in the data throughout the entire range. As will be seen later, however, this equation plays an important part in the present investigation.

9 Physicists have repeatedly attempted to find a relation between the pressures of different vapors at the same temperature such that a determination of the function $p = f(t)$ for one vapor would serve to determine this function for all other vapors. Ramsey and Young proposed the law

$$R = R' + k\ (T - T')$$

where R and R' are the ratios of the temperatures of two saturated vapors at two different pressures, and T and T' are the temperatures of one of the vapors corresponding to these pressures. They show that the law holds very closely for some 22 different substances arranged in 23 different pairs. In their paper they use the T and T' in the term $k\ (T - T')$ as being indiscriminately the numerators or the denominators of the ratios R and R'; i. e. they use the two equations

$$\frac{T_b}{T_a} = \frac{T'_b}{T'_a} + k'\ (T_a - T'_a) \qquad [3]$$

$$\frac{T_a}{T_b} = \frac{T'_a}{T'_b} + k\ \ (T_a - T'_a) \qquad [4]$$

without making any mention of the change from one form to the

other and without recognizing that the two forms lead to different results.

10 This law has been corroborated by the work of Richardson and by the work of Ramsey and Travers on crypton, argon and xenon.

11 Ayrton and Perry, and Everett have each remarked upon the lack of symmetry of equation [3] and have shown that equation [4] is symmetrical. This is most clearly demonstrated by Moss, who shows that [3] may be thrown into the form

$$\frac{T_b}{T_a}=\left[\frac{T_b'}{T_a'}-k'T_a'\right]+k'T_a$$

and [4] may be thrown into the form

$$\frac{T_a}{T_b}=\left[\frac{T_a'}{T_b'}-kT_a'\right]+kT_a$$

since T_a' and T_b' are the temperatures corresponding to some particular vapor pressure, these equations may be written respectively

$$\frac{T_b}{T_a}=k'T_a+c' \qquad [5]$$

and

$$\frac{T_a}{T_b}=kT_a+c \qquad [6]$$

12 Equation [6] may be written in the simple form

$$\frac{1}{T_b}=c\,\frac{1}{T_a}+k \qquad [7]$$

$\frac{1}{T_b}$ and $\frac{1}{T_a}$ thus being linear functions of each other.

13 Since equation [7] may be written

$$A-\frac{B}{T_b}=A'-\frac{B'}{T_a}$$

it follows that any equation to be applicable to all vapors in the same form with only its constants changed, and at the same time to be consistent with the temperature ratio law as stated above, must satisfy the condition that

$$p=f\left(A-\frac{B}{T}\right)$$

all other constants remaining the same for all vapors. Now, it happens that the equations which may be thrown into this form.

such as the Rankine short form, the Roche equation, etc., do not satisfy the experimental data for steam throughout its range with the necessary degree of accuracy. On the other hand, the Marks equation, which does satisfy these data with remarkable accuracy throughout the complete range, cannot be thrown into the required form. In view of the various considerations mentioned it has been decided in the present investigation to accept the temperature ratio law as expressed in equation [7], to use water as the standard substance, to use the Marks equation, equation [2], as representing the pressure-temperature relation for water, and to make a step-by-step solution of the pressure-temperature relation for ammonia by means of these equations.

14 The method used in applying the temperature ratio law and in determining the value of the constants c and k in equation [7], is that used by Moss, who has applied it to 17 different vapors. A description of this method and of the construction of Fig. 1 is given in Appendix No. 1.

15 After plotting the points representing the experimental data the straight line shown in Fig. 1 was drawn in a manner such as to represent all the points in the best possible manner.

16 The effectiveness of any method that enables one to throw any given data into a form such that it may be represented by a straight line lies in the fact that it discloses immediately and strikingly any departure from the general trend and reveals unerringly points that depart from this trend. It enables one to give various observations their proper weight and it affords a much safer basis of extrapolation than can be obtained from curves. The case in hand illustrates this principle. If the points are plotted on the regular pt plane it is easily seen that no smooth curve such as represents a law of nature, could, if passed through Regnault's lower temperature points and Brill's lower points, at the same time pass through Brill's points in the region -80 to -30 deg. fahr. Further than this the chart discloses nothing, as an infinite number of curves could be drawn, giving different weights to the different points, or all points could be given equal weight and the equation could then be determined by least squares. An inspection of Fig. 1 shows, however, that the straight line which best represents the whole range represents very accurately Regnault's lower points, Brill's highest point and all of his points from -80 to -110 deg. fahr. In the range -80 to -30 deg. fahr. all of Brill's points lie above the line, as does Davies' -41.8 deg.

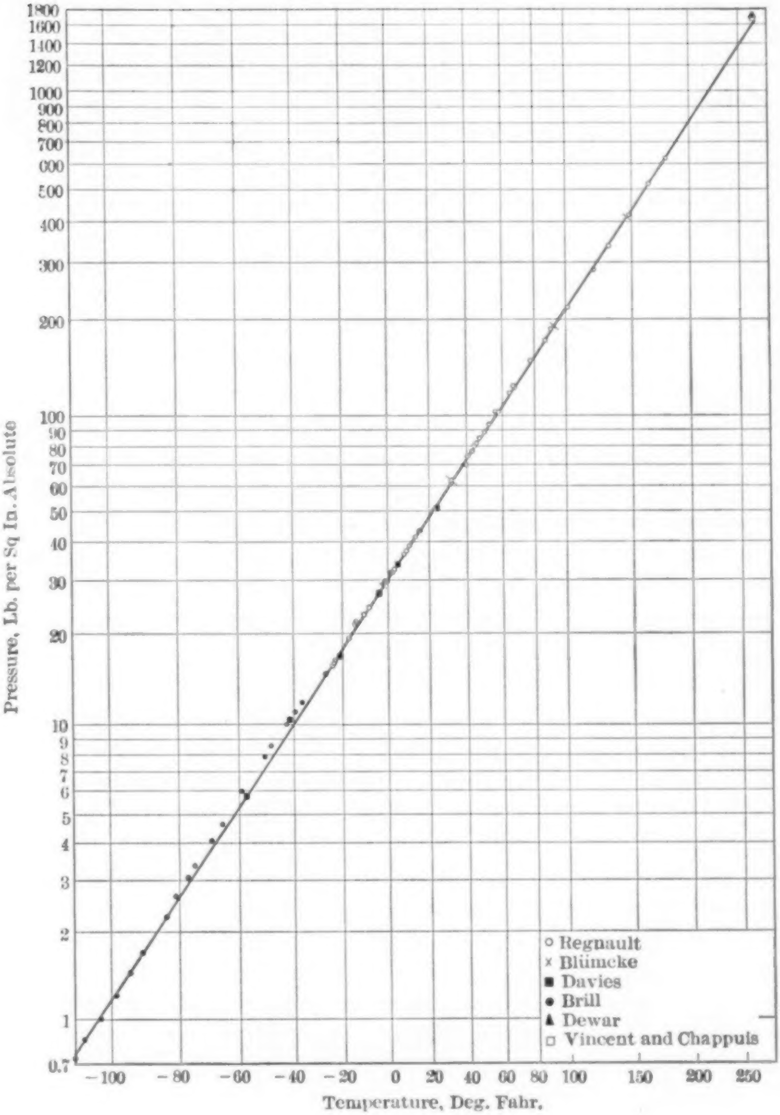

Fig. 1 Curve showing Agreement of Pressures found by Temperature Ratio Law with Those found by Experiment

fahr. point. It is significant, however, that Davies' −57.64 deg. fahr. point lies exactly on the line at precisely the point where Brill's points lie farthest from it.

17 Through the range −30 to +100 deg. fahr. the line represents the experimental points very accurately; above 100 deg. fahr. Regnault's points lie slightly below the line. Now all of Regnault's points above 100 deg. fahr. belong to his third series of experiments. In all of the higher pressure points of this series Regnault used a closed manometer and extensive corrections had to be applied to the height of mercury observed, and a comparison of the experiments in the range 40 to 90 deg. fahr. where the second and third series overlap shows that invariably the points determined in the third series by the use of the closed manometer lie below those determined in the second series by the use of the open manometer. The conclusion is evident that there was probably an error in the correction applied by Regnault to his readings with the closed manometer; the fact that these points lie below the line should therefore not be taken as conclusive evidence of the incorrectness of the line.

18 As to the critical data, it is seen that the points of both Dewar, and Vincent and Chappuis lie above the line. If the line be taken as correct this indicates either higher temperature, lower pressure, or both, at the critical point.

19 The same is the case with the determination of the critical data for water; as methods have been improved and greater precautions taken in the experimental work, the value found for the critical temperature has steadily risen until the latest determination, that of Holborn and Bauman, is 30 deg. higher than that found by Nadejdine 25 years earlier. It is reasonable to assume that the same errors may have been made in the ammonia determinations and that the true critical temperature of ammonia is higher than that found by either Dewar or Vincent and Chappuis. In the present investigation the critical data of ammonia are not of prime importance and the purely arbitrary assumption has been made, on the basis of the above reasoning and for want of better data, that the higher of the critical pressure determinations, that of Dewar, is correct. The corresponding temperature as determined by the temperature ratio law has been taken as the critical temperature. The resulting values are

$$p_k = 1690 \text{ lb. per sq. in.}$$

$$t_k = 273.2 \text{ deg. fahr.}$$

20 Since the above conclusions were drawn the articles giving the determinations of Jaquerod and Scheffer have been found. If plotted in Fig. 1 the point representing Jaquerod's determination falls below the straight line and that representing Scheffer's value falls almost exactly upon it. If Scheffer's value for the critical temperature is substituted in the temperature ratio law the resulting pressure is 1638.6 lb. per sq. in. while his experimental value is 1635.7 lb. per sq. in., the difference being about 1/6 of one per cent. These results offer corroboration of the correctness of the location of the straight line in Fig. 1, although the arbitrary values chosen for the critical temperature and pressure may be too high.

21 The equation of the straight line in Fig. 1 is

$$\frac{1}{T_a} = 1.70356 \frac{1}{T_w} - 0.0002242 \qquad [8]$$

In constructing tables with pressure as the argument the temperature of saturated steam at any pressure is found from steam tables. The corresponding saturation temperature of ammonia is then found by using equation [8] in the form

$$T_a = \frac{1}{\dfrac{1.70356}{T_w} - 0.0002242}$$

22 If temperature is taken as the argument the temperature at which steam will be saturated under the same pressure may be found by using equation [8] in the form

$$T_w = \frac{1}{\dfrac{0.587006}{T_a} + 0.0001316}$$

and the corresponding pressure is found from steam tables.

23 *Specific Volume of the Liquid.* There are available three sets of experimental data from which the specific volume of the liquid may be determined. Lange determined the density of liquid ammonia over the temperature range of −56 to +208 deg. fahr.; Dieterici, working by Young's method, obtained simultaneously the specific volumes of the liquid and of the saturated vapor over a temperature range of 32 to 222 deg. fahr. In addition to these there are the specific gravity determinations of d'Andréeff, from which the specific volume may be calculated; these experiments cover the range of 14 to 68 deg. fahr.

24 In Fig. 2 the experimental values are plotted with specific volumes as ordinates, and temperatures as abscissae. The form of equation used to express the volume of the liquid in terms of the temperature is that used by Avenarius, or

$$v' = a - b \log (t_k - t) \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

The curve shown in Fig. 2 up to 160 deg. fahr. represents this equation with the constants as follows:

$a = 0.06335$
$b = 0.016$
$t_k = 273.2$

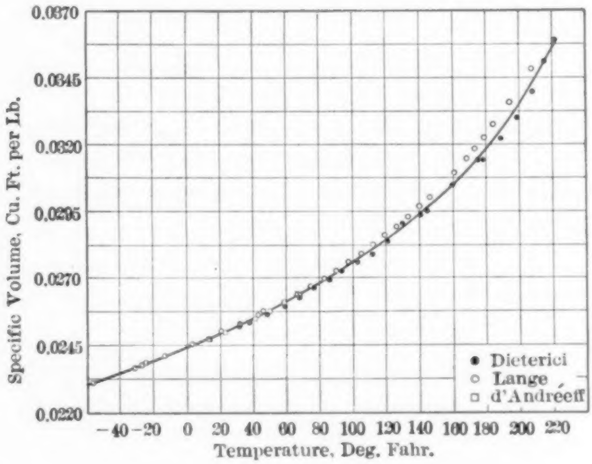

FIG. 2 CURVE SHOWING AGREEMENT OF VOLUMES OF LIQUID AS CALCULATED WITH THOSE FOUND BY EXPERIMENT

25 In determining these constants the work of Dieterici was given the most weight, both because the methods he employed are superior to the others, and because the volumes of the vapor to be used with these values are those determined by Dieterici by the same method in the same series of experiments.

26 The part of the curve above 160 deg. fahr. represents the values found by the method discussed in Paragraphs 45 to 50.

27 *Latent Heat of Vaporization.* Since the experimental information regarding the latent heat of vaporization of ammonia is too meager to be used as a basis for the determination of the relation existing between it and the temperature of vaporization, this

relationship will be determined from other considerations and the available data used as a check on the method.

28 Let [7] be differentiated with respect to the pressure. Then

$$\frac{-\dfrac{dT_a}{dp}}{T_a^2}=\frac{-c\dfrac{dT_w}{dp}}{T_w^2}$$

which may be written

$$\frac{T_a\dfrac{dp}{dT_a}}{T_w\dfrac{dp}{dT_w}}=\frac{1}{c}\cdot\frac{T_w}{T_a}$$

or since

$$\frac{T_w}{T_a}=c+k\,T_w$$

there results

$$\frac{T_a\dfrac{dp}{dT_a}}{T_w\dfrac{dp}{dT_w}}=1+\left(\frac{k}{c}\right)T_w \qquad [10]$$

Now by the Clapeyron relation connecting the latent heat and the absolute temperature of vaporization with the change of volume during vaporization

$$\left(\frac{v''-v'}{r}\right)=\frac{1}{144AT\left(\dfrac{dp}{dT}\right)_{sat}}$$

Substituting in equation [10]

$$\frac{\dfrac{(v''-v')_w}{r_w}}{\dfrac{(v''-v')_a}{r_a}}=1+\frac{k}{c}T_w \qquad [11]$$

or with the proper values of the constants introduced

$$\frac{\dfrac{(v''-v')_w}{r_w}}{\dfrac{(v''-v')_a}{r_a}}=1-0.0001316T_w \ldots\ldots\ldots\ldots [12]$$

29 The quantity $\dfrac{(v''-v')_w}{r_w}$ may be found from steam tables or as follows: From the Clapeyron relation

$$\frac{(v''-v')_w}{r_w}=\frac{1}{144\,AT\left[\dfrac{dp}{dT_w}\right]_{sat}}$$

By differentiating [2]

$$\left[\frac{dp}{dT_w}\right]_{sat}=p\left[\frac{11222.13}{T_w^2}+0.00000641484\;T_w-0.00932768\;T_w^2\right]$$

Substituting this value in the Clapeyron relation

$$\frac{(v''-v')_w}{r_w}=\frac{1}{p\left[\dfrac{2078.07}{T_w}-0.00172726\;T_w+0.00000118787\;T_w^2\right]} \quad [13]$$

30 By the use of equations [12] and [13] $\dfrac{(v''-v')a}{r_a}$ may be calculated and if either numerator or denominator is known the other may be found. Fortunately there are available the experimental determinations of Dieterici of the specific volumes of the liquid and saturated vapor; the former were given in Pars. 23–26 and the latter will be discussed in Pars. 41–44. From these determinations various values of $(v''-v')$ were found and values of r calculated as described above.

31 According to the generally accepted ideas concerning the critical point, at that point $r=0$ and $\dfrac{dr}{dt}=-\infty$. These facts led Thiesen to suggest as an empirical formula

$$r=c\;(t_k-t)^n$$

This form of equation has been used for water by Thiesen, Henning, and Marks and Davis, and for ammonia by Dieterici. When plotted

on logarithmic cross-section paper it is represented by a straight line, the slope being equal to n.

32 The values of r found by the above method were plotted to the corresponding values of (t_k-t) on logarithmic cross-section paper and were found to lie almost exactly on a straight line, indicating that the Thiesen formula might be used to express the desired relation. The formula may be written in the form

$$\log r = \log C + n \log (t_k - t)$$

and it was found that the values of the constants giving the best agreement with the plotted points are

$$\log C = 1.856064$$

and

$$n = 0.37$$

Therefore the final equation is

$$\log r = 1.856064 + 0.37 \log (273.2 - t) \ldots\ldots\ldots\ldots [14]$$

33 The available data concerning the latent heat of vaporization are given in Table 1 and the various points are plotted in Fig. 3, the full line in this figure representing equation [14]. Of the determination of r made by Regnault, the greater number were lost, but 12 were saved and later published. The results of these experiments do not give r directly and have been variously interpreted by different writers. The table contains the interpretation of Jacobus and the three values of r as quoted from Regnault by Dieterici. Two values for r are given by von Strombeck, one being an average of 12 experiments, the other of eight. The value given by Franklin and Kraus is an average of three determinations at the normal boiling point, and the value obtained is exactly the value deduced by the same writers from the absolute boiling point and the molecular elevation. A different value was found at the normal boiling point by Estreicher and Schnerr. The values given by Denton and Jacobus, represented in Fig. 3 by crosses, were calculated from readings taken during a test of an ammonia compressor, and hence cannot be considered of much weight; because of the scarcity of scientific data, however, these points are included in the table and chart.

34 In Fig. 3 are also plotted the curves that represent various equations used in calculating the latent heat of vaporization, as given in the various existing tables. These are plotted exactly as given and it should be remembered that there is some variation in the value of the heat unit used by different writers; consequently the

various curves are not exactly comparable and are given simply to show the variation in the values of latent heat as given in the tables now in use.

TABLE 1 SUMMARY OF VARIOUS DETERMINATIONS OF THE LATENT HEAT OF VAPORIZATION

Temperature, Deg. Cent.	Cal. per Kg.	Temperature, Deg. Fahr.	B.t.u.	Authority
10.90	287.0	51.62	516.6	Regnault (Jacobus)
15.53	285.2	59.95	513.3	
16.00	290.5	60.80	522.8	
12.94	283.8	55.29	510.8	
11.90	285.8	53.42	514.4	
10.73	288.1	51.31	518.5	
11.04	292.5	51.87	526.4	
10.15	292.4	50.27	526.2	
9.52	295.0	49.14	531.0	
10.99	293.3	51.78	527.9	
12.60	291.6	54.68	524.8	
7.80	291.8	46.04	525.2	
7.80	294.2	46.04	529.5	Regnault (Dieterici)
11.00	291.3	51.80	524.3	
16.00	297.4	60.80	535.3	
19.53	296.5	67.15	533.7	Von Strombeck
17.00	296.8	62.60	534.2	
−33.64	337.0	−28.55	606.6	Franklin and Kraus
−33.40	321.3	−28.12	578.3	Estreicher and Schnerr
.....	.....	84.6	524.8	Denton and Jacobus
.....	.....	82.7	525.7	
.....	.....	87.7	512.4	
.....	.....	−10.7	569.2	
.....	.....	− 3.2	603.5	
.....	.....	+14.5	570.4	

35 The equations of the curves shown in Fig. 3 are

Ledoux .... $r=583.33-0.5499t-0.0001173t^2$
Wood .... $r=555.5-0.613t-0.000219t^2$
Peabody .... $r=540-0.8\ (t-32)$
Dieterici .... $\log r=1.56141+0.5 \log (266.9-t)$
Goodenough .... $\log r=1.7920+0.4 \log (266-t)$

36 No comment is necessary regarding the curves for r according to Ledoux, Wood and Peabody, since their equations are not of a form such as to give correct results at high temperatures. Moreover they were derived before there were experimental determinations of the volume of the saturated vapor, from which latent heats could be calculated.

37 Dieterici used Zeuner's empirical expression for the second term of Regnault's pressure-temperature equation, and omitted the third term entirely. This approximation is quite accurate for low temperatures, but as the temperature increases the pressures thus calculated become too small, and this error increases rapidly at high temperatures. For instance, Regnault extrapolates his curve and gives for 212 deg. fahr. a pressure of 901.6 lb. per sq. in., while the

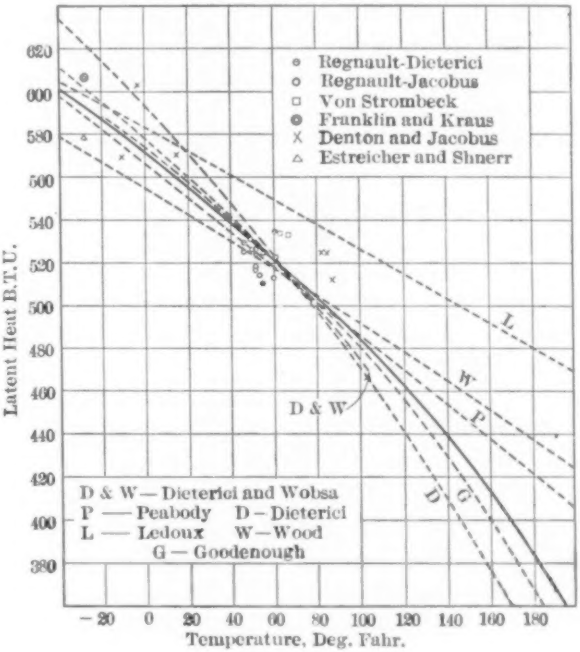

Fig. 3 Curves showing Comparison of Experimental Values for Latent Heat of Vaporization with Values as calculated by Various Writers

value calculated by Dieterici for this temperature is 847.1 lb. per sq. in. or 6 per cent lower. As the error is increasing, the values of $\left[\frac{dp}{dt}\right]_{sat}$ are too small and values of r calculated by means of the Clapeyron equation are too small at high temperatures. Since the relation employed between r and $(t_k - t)$ is a straight line on logarithmic paper and the values of $\frac{dp}{dt}$, consequently of r, agree at some

medium temperature, the values given by the equation at low temperatures will be too high. This is shown in Fig. 3, where it is seen that Dieterici's curve passes above even the highest of the actual experimental determinations of r at -28 deg. fahr.

38 The equation used by Goodenough fits the actual determinations and is consistent with the Clausius relation, using Dieterici's values for volumes and Goodenough's constants in Bertrand's pressure-temperature equation. This latter equation, however, applies over but a small temperature range. The variation between Goodenough's r curve and the present one is due to the different pressure-temperature relation used and the higher value assumed for the critical temperature.

39 It is believed that, due to the form of equation [14] and the accuracy with which it represents the derived values of r where known, the equation may be safely extrapolated as far as required in this investigation.

40 The comparison between the values of r given by equation [14], the corresponding values of v'' derived from the Clausius relation, and the experimental values for v'', properly belongs in the next section and is there shown.

41 *Specific Volume of Saturated Vapor.* The only available experimental determinations of the specific volume of the saturated vapor are those of Dieterici. These experiments extend up to a temperature of 222 deg. fahr., but unfortunately were not carried below a temperature of 32 deg. fahr. Owing to the form of the curve representing the volume-temperature relation, any extended extrapolation of the curve or of any empirical equation to represent this relation would be very unsafe below 32 deg. fahr., where the volume is seen to increase very rapidly.

42 As shown in the preceding section, however, equation [14] is believed to be of a form such that extrapolation can be carried to very low temperatures. Therefore in the present investigation the volumes of the saturated vapor have been calculated as follows:

43 By the use of equations [12] and [13] values of $\dfrac{(v''-v')_a}{r_a}$ were calculated. These values multiplied by the corresponding values of r found from equation [14] gave the values of $(v''-v')$. The addition of the proper values of v' found from equation [9] gave the values of v'', the quantity desired.

44 The agreement of the values of v'' calculated by the above

method is shown in Fig. 4, the curve in this figure representing the calculated values and the points the observed values.

45 *Specific Volume of Liquid and Saturated Vapor at High Temperatures.* The "law of the straight diameter" was first proposed by

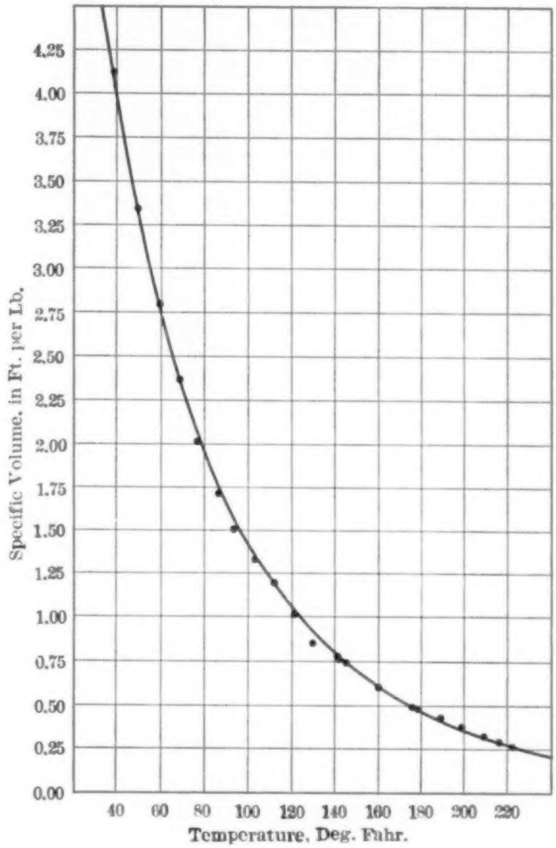

Fig. 4 Curve showing Agreement of Volumes of Saturated Vapor as calculated with Those found Experimentally by Dieterici

Cailletet and Mathias in 1886. This law, as originally given, stated that if the densities of a liquid and its saturated vapor are plotted (as abscissae) against the corresponding temperature (as ordinates) to form a dome, the mid-points of the horizontal chords of this dome will lie in a straight line nearly parallel to the axis of temperatures. References to the law may be found in an article by Davis.[1]

[1] Proc. Am. Acad. of Arts and Sciences, vol. 45, 1910, p. 305.

TABLE 2 THE LAW OF THE STRAIGHT DIAMETER FOR AMMONIA

Temperature, Deg. Fahr.	Density of Vapor	Density of Liquid	Mean Density	By Formula	Difference
−40	0.0393	42.704	21.372	21.372	±0.000
−30	0.0517	42.296	21.174	21.174	±0.000
−20	0.0671	41.883	20.975	20.974	−0.001
−10	0.0860	41.461	20.774	20.774	±0.000
0	0.1088	41.039	20.574	20.572	−0.002
+10	0.1362	40.608	20.372	20.370	−0.002
20	0.1689	40.169	20.169	20.167	−0.002
30	0.2075	39.722	19.965	19.964	−0.001
40	0.2525	39.267	19.760	19.759	−0.001
50	0.3051	38.803	19.554	19.554	±0.000
60	0.3657	38.329	19.347	19.347	±0.000
70	0.4355	37.844	19.140	19.140	±0.000
80	0.5152	37.348	18.932	18.932	±0.000
90	0.6061	36.841	18.724	18.724	±0.000
100	0.7099	36.319	18.514	18.514	±0.000
110	0.8265	35.782	18.304	18.303	−0.001
120	0.9590	35.228	18.093	18.092	−0.001
130	1.1062	34.655	17.881	17.880	−0.001
140	1.2756	34.061	17.668	17.667	−0.001
150	1.4643	33.444	17.454	17.453	−0.001
160	1.6763	32.798	17.237	17.238	+0.001

46 It was found by Young that the diameter is actually straight in the case of all but a few substances, normal pentane being an example of this class. In the case of most substances, however, the diameter can be represented accurately by a second degree equation; some substances, such as alcohols, require a third degree equation.

47 In order to find the equation of the straight diameter for ammonia the values of the densities of the liquid and of the saturated vapor were found at 10 deg. intervals. The values for the liquid were calculated by the use of equation [9], and those for the vapor by the method described in Pars. 41–44. At each of the temperatures the mean density was calculated. The results are given in Table 2 and plotted in Fig. 5, where the points for the liquid and vapor are shown as large circles and the mean points as small circles. The diameter is seen to be slightly curved, but it may be accurately represented by the second degree equation

$$\gamma = 19.3473 - 0.02067\ (t-60) - 0.0000042\ (t-60)^2 \ldots\ldots [15]$$

The table shows that this equation represents the mean densities as calculated by the other method with a maximum error of about

1/100 of one per cent over a temperature range of -40 to $+160$ deg. fahr.

48 The method used at ordinary temperatures for finding the change of volume on vaporization, or $(v''-v')$, may be employed at temperatures up to the critical point if it is assumed that the values

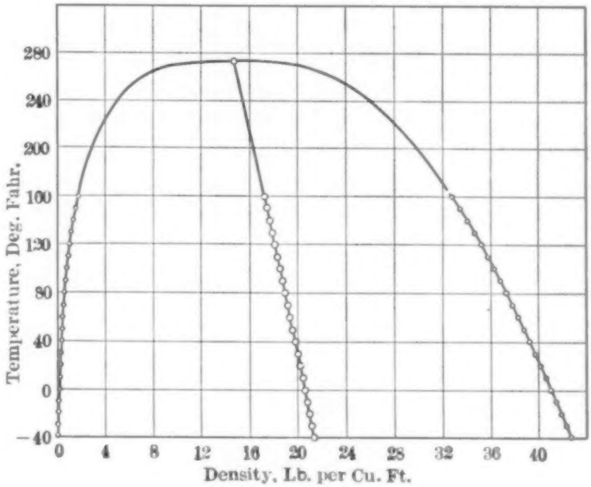

FIG. 5 CURVE SHOWING DOME OF TEMPERATURE-DENSITY PLANE AND "STRAIGHT DIAMETER" OF CAILLETET AND MATHIAS

for r found by equation [14] are correct up to that point. Equation [15] gives values for

$$\frac{\gamma''+\gamma'}{2}=\frac{1}{2}\left[\frac{1}{v''}+\frac{1}{v'}\right]$$

A simple algebraic manipulation gives

$$v''=\frac{\gamma(v''-v')+1+\sqrt{\gamma^2(v''-v')^2+1}}{2\gamma}$$

Having values for v'' and $(v''-v')$, the values for v' may be easily found. This method enables the values of the volumes of the liquid and the saturated vapor to be found at least qualitatively up to the critical point itself.

49 Above 160 deg. fahr. the values found for v' by this method were materially lower than those found by equation [9] and that

the difference increased as the temperature increased above this point. An inspection of equation [9] shows that this equation would give a value of $+\infty$ for the volume of the liquid at the critical point, whereas at this point the liquid has a finite volume. In the case of other vapors it has been shown that the form of the equation used, that of Avenarius, gives accurate values at temperatures somewhat removed from the critical point, but that it does not hold near this point. Therefore, in the present case equation [9] has been used only up to 160 deg. fahr., and the values above this temperature have been calculated by the law of the straight diameter.

50 Since the diameter is nearly parallel to the axis of temperatures a considerable error in the value of the critical temperature will cause but a small error in the resulting value of the critical density. The substitution in equation [15] of the assumed value of t_k, or 273.2 deg. fahr., gives for the critical density and volume respectively

$$\gamma_k = 14.75 \text{ lb. per cu. ft.}$$
$$v_k = 0.0678 \text{ cu. ft. per lb.}$$

51 *Specific Volume of the Superheated Vapor.* The attempt has often been made to deduce rationally an equation of state which with suitable change of constants will represent the relation $f(P, v, T)=0$ for various fluids in all states from the gaseous condition above the critical temperature to the liquid condition. Such equations are constructed with special reference to the behavior of fluids in the neighborhood of the critical state and apply more particularly to fluids, the critical temperature of which is within the range encountered in practice. In the case of a fluid such as ammonia, however, the critical temperature of which is far above the working range, purely empirical equations of simpler form give better results throughout the small range covered in practice, and in addition lend themselves much more readily to the formation of the various heat equations. Of these equations those of Ledoux, Zeuner, Peabody and Wood were based upon various doubtful assumptions, while Wobsa's first equation gives values that do not closely agree with experimental results, and only his second equation

$$v-a=\frac{BT}{p}-\frac{c}{T^n}+\frac{b}{p} \qquad [16]$$

with

$B = 49.736$
$a = 0.0075$
$C = 2450$
$n = 2$
$b = 80$

is quite satisfactory in this respect.

52 In choosing a characteristic equation several points must be considered.

a The equation must represent with fair accuracy the available reliable experimental data on the relations of p, v and T.

b The equation should be of a form such as to make the various thermodynamic relations derived from it as simple as possible.

c These derived equations must represent accurately the experimental data.

53 Wobsa's second equation fulfils the first of these conditions admirably. It is, however, somewhat defective with respect to the other requirements. The good results obtained from Goodenough's equation for superheated steam

$$v+c=\frac{BT}{p}-(1+ap)\frac{m}{T^n} \quad \text{[17]}$$

suggested the adoption of the same form of equation for superheated ammonia in the preliminary investigation. This equation satisfies the second condition in that it gives derived relations of comparatively simple form. A trial with various sets of constants showed that it could be made to represent the volume measurements substantially with the same accuracy as Wobsa's second equation, and thereby satisfy the first condition. Having established the fact that the proposed equation is permissible, the next step was the determination of the constants. In connection with this process emphasis must be placed on the third consideration heretofore mentioned. From the characteristic equation are derived expressions for (*a*) the specific heat at constant pressure; (*b*) the heat content of the superheated and also of the saturated vapor; (*c*) the Joule-Thompson coefficient. Hence the constants must be chosen not with reference to volume alone. While the volume measurements must be satisfied,

three other derived relations must conform equally well to the experimental data in the respective fields.

54 With due consideration of all the conditions the following values were finally assumed for the constants:

$B = 0.6321$, p in lb. per sq. in.
$\log m = 12.900000$
$c = 0.100$
$n = 5$
$a = 0$

The final equation with constants inserted is therefore

$$v + 0.100 = 0.6321\frac{T}{p} - \frac{79433 \times 10^{8}}{T^{5}} \quad \text{[18]}$$

55 It will be seen that the constant a is taken as zero. In view of the fact that the available data consist only of values along the saturation curve and in most cases but one point on each isotherm in the superheated region, no information is available regarding the shape or curvature of these isotherms on the $pv-p$ plane; moreover the derived equations demand an exceedingly small value of a. The use of straight lines for these isotherms therefore seems to be as well justified as the use of parabolas; hence a was made equal to zero.

56 A summary of the various direct determinations of the specific volume of the superheated vapor according to Perman and Davies and LeDuc and Guye is given in Table 3. In Table 4 the values given by equation [18] and by Wobsa's first and second equations are compared with these experimental determinations. It is seen that both equation [18] and Wobsa's second equation give results agreeing better with experiment at the one atmosphere points than those obtained from Wobsa's first equation. At the 32, 122 and 212 deg. fahr. points at one atmosphere pressure equation [18] and Wobsa's second equation give practically the same percentage deviation from the experimental values. At -4 deg. Wobsa's second equation gives a better agreement with the experimental value than does equation [18]; this point, however, is considerably below the range found for superheated ammonia in practice, and it is believed that the considerations appearing in the discussion of the heat equations derived from the characteristic equation justify the use of equation [18] in preference to Wobsa's equations.

57 In Fig. 6 the isotherms deduced from equation [18] are drawn on the $pv-p$ plane and the various experimental points are plotted

in order to show the agreement. From this chart it is seen that if both of the determinations of Perman and Davies at 32 deg. are correct, the one at one atmosphere and the other at one-half atmosphere, then either the saturation curve should lie considerably higher or the 32 deg. isotherm should be sharply curved. Since the one atmosphere

TABLE 3 SUMMARY OF VARIOUS DETERMINATIONS OF VOLUME OF THE SUPERHEATED VAPOR

Pressure, Atmos.	Temperature, Deg. Cent.	Volume in Liters per Gram	Temperature, Deg. Fahr.	Volume, Cu. Ft. per Lb.	Authority
1/2	0	2.6096	32	41.8006	Perman
1	−20	1.19575	−4	19.1535	Perman
	0	1.2973	32	20.7802	Perman
	50	1.5473	122	24.7847	Perman
	100	1.7964	212	28.7747	Perman
	0	1.2955	32	20.7513	Le Duc
	0	1.2974	32	20.7818	Guye

TABLE 4 COMPARISON OF VALUES OBTAINED FROM VARIOUS EQUATIONS FOR VOLUME OF SUPERHEATED VAPOR WITH EXPERIMENTAL VALUES

Pressure, Atmos.	Temperature, Deg. Fahr.	Experimental Values		Computed from 18		Computed by Wobsa I		Computed by Wobsa II	
		Authority	Value	Value	% Diff. from Exp.	Value	% Diff. from Exp.	Value	% Diff. from Exp.
1	−4	Perman	19.154	19.093	−0.32	19.121	−0.17	19.140	−0.07
	32	Perman	20.780	20.769	−0.05	20.745	−0.17	20.767	−0.06
		Guye	20.782	20.769	−0.06	20.745	−0.18	20.767	−0.07
		Le Duc	20.751	20.769	+0.09	20.745	−0.03	20.767	+0.08
	122	Perman	24.785	24.798	+0.05	24.746	−0.16	24.775	−0.004
	212	Perman	28.775	28.730	−0.16	28.695	−0.28	28.725	−0.17
1/2	32	Perman	41.801	41.916	+0.28	41.821	+0.05	41.939	+0.33

determination agrees closely with the determinations of LeDuc and Guye, since there is good authority for the location of the saturation curve, and since the flatness of the isotherms is fairly well established, it would seem that the half atmosphere point of Perman and Davies is probably in error.

58 In Fig. 6 the full line curve represents values resulting from equation [18] and the dotted line curve represents the values resulting from equation [15]. A comparison of Wobsa's equations is not given because the constants in those equations were determined on the assumption of a different pressure-temperature relation along the saturation curve; the substitution in Wobsa's equations of the

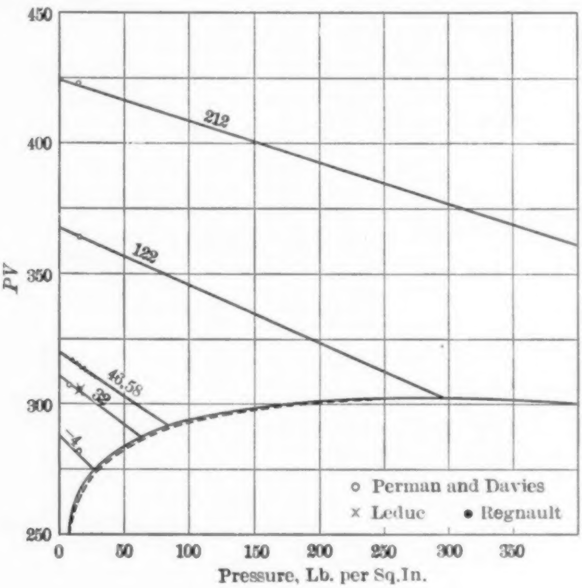

FIG. 6 COMPARISON OF ISOTHERMS DEDUCED FROM EQUATION [18] WITH POINTS REPRESENTING EXPERIMENTAL DETERMINATIONS SHOWN ON $PV-P$ PLANE

values of pressure corresponding to given temperatures as used in this investigation would not therefore give values of volume comparable with those deduced from equation [18]. It may be stated, however, that Wobsa's equations give very fair agreement with the values found experimentally by Dieterici, the maximum deviation of the second equation from the values given in Dieterici's table being a little over 2 per cent.

59 In addition to the preceding the only data available for the purpose of checking the characteristic equation are the results of a series of experiments performed by Regnault. In these experiments measurements were made of the relative volumes occupied by a

quantity of ammonia gas at different pressures along the isotherm corresponding to a temperature of 8.1 deg. cent. or 46.58 deg. fahr. As the weight of ammonia used was not recorded, the experiments can only be used to obtain the relative values of the specific volume or of the product pv along this isotherm. In making use of these values the products pv were plotted against the corresponding values of p, both quantities being measured in the units employed by Regnault. A straight line was next passed through these points by the method of least squares. It was found that the pressure equivalent to 20 lb. per sq. in. came at about the center of the group of points, and the value of the specific volume was calculated by equation [18] for this pressure and a temperature of 46.58 deg. fahr.; this value multiplied by the pressure gave a value of the product pv in English units. The value of this product in the units used by Regnault was then found at the point where the straight line determined by least squares crossed the pressure coördinate equivalent to a pressure of 20 lb. per sq. in. From these two values for pv the conversion factor 0.00561 was found, this being the number by which Regnault's values of pv must be multiplied in order that the isotherm calculated from equation [18] shall pass through the center of Regnault's group of points. All of his values of pv were multiplied by this factor and the points plotted in Fig. 6 as small black dots. The only information they afford is a check on the slope of the isotherm for 46.58 deg. fahr. on the $pv-p$ plane. The points seem to indicate that the isotherms near this temperature at least, are very flat curves or even straight lines.

60 *Specific Heat of the Superheated Vapor.* The first determination of the specific heat of superheated ammonia was made by Regnault, who found a value of 0.50836 at atmospheric pressure and over a temperature range of 75 deg. fahr. to 420 deg. fahr. Later Wiedemann performed two series of experiments at a pressure of about 16 lb. per sq. in.; he found a value of 0.5202 between 77 and 212 deg. fahr. and a value of 0.5356 between 77 and 392 deg. fahr. He proposed the equation

$$c_p=0.4949+0.000172t$$

61 Thus far the history is much the same as in the case of superheated steam, the specific heat of which was long supposed to be independent of both temperature and pressure and equal to 0.48 as determined by Regnault; later the investigations of Maillard and LeChatelier and of Lange agreed in making it a linear function of

the temperature. In the case of steam, however, the experiments of Knoblauch and Jakob, Thomas, and Knoblauch and Mollier have shown that the specific heat of superheated steam depends upon the pressure. While no experiments have been made to investigate the variation of the specific heat of superheated ammonia with pressure, it is very probable that there is such a variation and its rate may be determined by means of Clausius' thermodynamic relation providing a sufficiently accurate characteristic equation has been determined. The process of obtaining an explicit expression for c_p in terms of the variables p and T has inherent difficulties and is rendered more difficult in this case by the meagerness of the data available. The success with which the method has been applied to superheated steam by Goodenough, however, seems to warrant its use in the present investigation.

62 The application of the Clausius relation to equation [18] readily gives the following equation for c_p:

$$c_p = \phi\,(T) + \frac{Amn\,(n+1)}{T^{m+1}}\,p \quad \ldots\ldots\ldots\ldots\ldots [19]$$

For the complete derivation of this equation see Appendix No. 2.

63 If the arbitrary function $\phi\,(T)$ can be determined, there will result an explicit expression for c_p in terms of the variables p and T. Wobsa has used the same method of analysis starting with his own characteristic equations and has obtained equations for c_p similar to the present one.

64 The determination of the proper function $\phi\,(T)$ is the critical point of the investigation. Wobsa equated his expression for c_p at atmospheric pressure to Wiedemann's linear equation for c_p at atmospheric pressure and thus found the $\phi\,(T)$ to be of the form

$$\phi\,(T) = a + bt - \frac{c}{T^n}$$

He shows that the terms in the third power of T, which is the correction term for atmospheric pressure, is vanishingly small above 212 deg. fahr. and that above this temperature c_p at infinitely small pressures will not differ from c_p at atmospheric pressure, or $\phi\,(T) = 0.4949 + 0.000172t$. Below 212 deg. fahr. where the correction term becomes appreciable, Wobsa substitutes the linear function $\phi\,(T) = 0.4712 + 0.000278t$.

65 For the determination of ϕ (T) in the present investigation there are available the recent determinations of c_p at high temperatures by Nernst. In addition to the results of his own work, Nernst gives a summary of the results of previous investigations (see Table 5). In this table the values are given as molecular specific heats and these have been converted into specific heats per unit of weight by dividing by the molecular weight of NH_3 or 17.064. It is seen that the values given for Wiedemann are lower than those quoted earlier in this section from the original; while the value given for Regnault is higher than the original. As the original papers of Keutel and

TABLE 5 SUMMARY OF C_p MEASUREMENTS AS GIVEN BY NERNST

Temperature, Deg. Cent.	Molecular C_p	Temperature, Deg. Fahr.	C_p	Authority
20	8.64	68	0.506	Keutel
20	8.62	68	0.505	Voller
25–100	8.84	77–212	0.518	E. Wiedemann
25–200	9.11	77–392	0.534	E. Wiedemann
24–216	8.71	75–421	0.510	Regnault
365–567	10.4	689–1053	0.609	Nernst
480–680	11.2	896–1256	0.656	Nernst

Voller are not at hand the values given in this table for their results are assumed to be correct, but the values used for Wiedemann and Regnault are those taken from the original.

66 If in the equation for c_p, $p=0$, the equation reduces to $c_p=\phi$ (T). Therefore if we have values for c_p at zero pressure the ϕ (T) may be determined. To obtain these values the proper correction terms were calculated and subtracted from the values of c_p. The resulting values of $(c_p)_o$ seemed to justify the assumption that $(c_p)_o$ is a linear function of the temperature. This linear relation may not, and probably does not, hold for the entire range of superheat, but it may be assumed as a close approximation, and is the only assumption justified by the experimental data available. The function therefore takes the simple form

$$\phi\ (T)=a+\beta T$$

and the equation for c_p becomes

$$c_p=a+\beta T+\frac{Amn\ (n+1)}{T^{n+1}}p \quad \ldots\ldots\ldots\ldots\ldots [20]$$

67 The constants m and n are those heretofore given in connection with the characteristic equation, and the values finally chosen for α and β are: $\alpha=0.382$; $\beta=0.000174$. Replacing $Amn\ (n+1)$ by a single constant C, and substituting the proper value for the various constants the formula for the specific heat becomes:

$$C_p=0.382+0.000174\ T+p\frac{C}{T^6}\ \dots\dots\dots\dots\dots [21]$$

where $\log C=13.644705$.

68 Values of c_p have been calculated for various pressures up to 400 lb. per sq. in. and for temperatures up to 1100 deg. fahr.; these values are plotted against temperatures in Fig. 7. The experimental determinations are also plotted to show the agreement. The determinations of Regnault, Nernst, Keutel and Voller were made at atmospheric pressure, while those of Wiedemann were at a pressure of about 16 lb. per sq. in. absolute. The agreement between the curve derived from equation [21] for a pressure of 15 lb. per sq. in. and the points representing the determinations of Regnault, Nernst, Keutel and Voller is seen to be very good, while the points representing Wiedemann's determinations lie considerably above the curve.

69 *Heat Content of Saturated and Superheated Vapor.* Having an explicit formula for the specific heat at constant pressure in terms of the variables p and T, an expression for the heat content in terms of these variables may be easily derived:

$$i=\alpha T+\frac{\beta}{2}T^2-A(n+1)p\frac{m}{T^n}-Acp+i_0\dots\dots\dots [22]$$

For complete derivation of this formula, see Appendix No. 3.

70 The constant of integration i_0 is determined by passing to the saturation limit. Since the constants α, β, m, n and c are known, the value of the right-hand member of equation [22], exclusive of i_0 may be found for any given pressure and temperature. If the value of i is known at this point, i_0 may be determined by subtraction. Now for the saturated vapor

$$i''=i'+r=u+Apv'+r$$

If the energy u be taken equal to zero at 32 deg. fahr., then at this temperature

$$i''_{32}=0+Ap_{32}v'_{32}+r_{32}$$

Substituting the proper values as determined by the formulae for the saturated vapor there results

$$i''_{32}=0.3+546.4=546.7$$

By the substitution of the proper values of p and T in equation [22] there results

$$i=i_o=188.7$$

By subtraction $i_o=358.0$

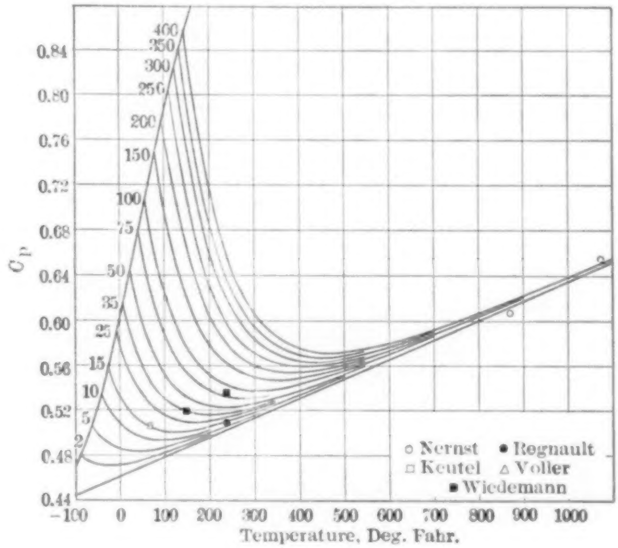

FIG. 7 CURVES GIVING VALUES OF SPECIFIC HEAT AT CONSTANT PRESSURE

Introducing known constants equation [22] becomes, with pressure in lb. per sq. in.

$$i=0.382T+0.000087T^2-p\,\frac{C}{T^5}-0.0185p+358.0\ldots\ldots[23]$$

where $\log C=12.945735$.

71 There are no direct measurements of the total heat or heat content of the superheated or saturated vapor by which equation [23] may be checked. A check on the accuracy of the equation at the saturation limit is afforded by a comparison of the values of the heat content of the liquid, obtained by subtracting values of r calculated by equation [14] from values of i'' calculated by equation

[23], with the experimental determinations of Dieterici. This comparison is shown in the following section. A check on the accuracy of the formula in the superheated region may be obtained by a comparison, by means of the law of corresponding states, of the values for the Joule-Thompson coefficient resulting from equation [23] with the values of this coefficient for other vapors. There is also available

TABLE 6 HEAT CONTENT OF LIQUID AS DEDUCED FROM EXPERIMENTS OF DIETERICI AND DREWES

C'im (t − 0° C)	u' Above 0°C. Cal. of 4.222 Joules	u' Above 0°C. Cal. of 4.184 Joules	Temperature, Deg. Cent.	Temperature, Deg. Fahr.	u' Above 32° F. in B.t.u.	$i' = u' + Apv'$	Authority
1.148	10.65	10.75	9.28	48.70	13.95	19.77	Dieterici
1.162	10.93	11.03	9.41	48.94	19.85	20.27	
1.141	24.25	24.47	21.25	70.25	44.04	44.67	
1.140	24.57	24.79	21.55	70.79	44.62	45.26	
1.139	38.67	39.02	33.95	93.11	70.23	71.21	
1.147	39.27	39.62	34.24	93.63	71.31	72.30	
1.200	42.03	42.41	35.02	95.03	76.34	77.35	
1.172	48.17	48.60	41.1	105.96	87.46	88.68	Drewes,—reliable according to Dieterici
1.158	48.35	48.78	41.75	107.15	87.78	89.02	
1.186	61.44	62.00	51.8	125.22	111.60	113.25	
1.163	60.42	60.97	51.95	125.46	109.75	111.41	
1.187	70.74	71.38	59.6	139.3	128.50	130.57	
1.164	72.75	73.41	62.5	144.5	132.15	134.40	
1.179	82.76	83.50	70.2	158.4	150.30	153.07	
1.205	87.48	88.27	72.6	162.7	158.90	161.84	
0.974	11.50	11.60	11.8	53.24	20.89	21.35	Drewes,—questioned by Dieterici
1.098	21.62	21.81	19.7	67.46	39.26	39.86	
1.069	22.82	23.02	21.35	70.43	41.44	42.08	
1.140	34.61	34.92	30.36	86.64	62.85	63.72	
1.148	35.83	36.16	31.21	88.17	65.08	65.97	
0.7065	−10.49	−10.59	−14.85	5.27	−19.06	−18.91	

a set of throttling experiments performed by Wobsa. This check is given in Appendix No. 4 and Fig. 10.

72 *Heat Content of the Liquid.* The values of heat content of the liquid may be found by subtracting the values of r found by equation [14] from the values of i'' found by equation [23]. The most important check on these values is derived from a series of experiments performed by Dieterici and by Drewes working in Dieterici's laboratory. In these experiments the quantity measured was the amount of heat given up by a certain weight of a mixture of saturated and liquid ammonia cooling at constant volume in a sealed tube.

This quantity was corrected for the heat due to the latent heat of the portion of vapor condensed and the remaining amount of heat was divided by the change in temperature to obtain the mean "inner" specific heat of the liquid over that temperature range. In his paper Dieterici gives only the formula which he chose as best representing the results of his experiments, which states that the mean "inner"

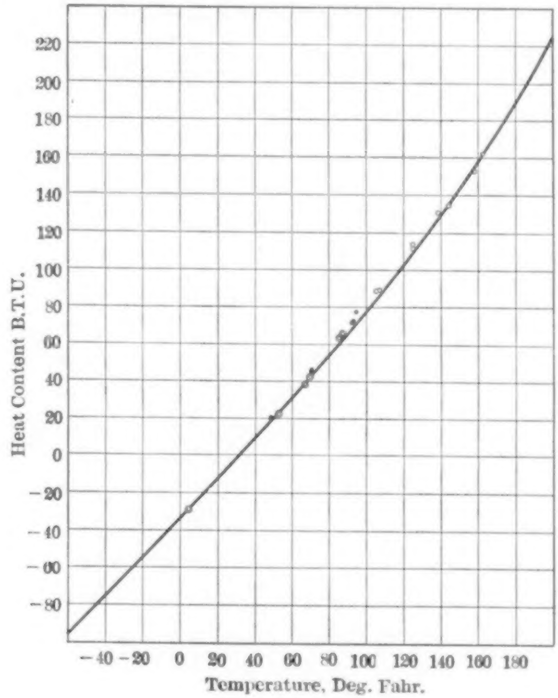

FIG. 8 COMPARISON OF CALCULATED VALUES OF HEAT CONTENT OF LIQUID WITH VALUES DEDUCED FROM EXPERIMENTAL WORK OF DIETERICI AND DREWES

specific heat between t and 32 deg. fahr. $=1.118+0.000578\,(t-32)$. Since this is a linear function of the temperature it follows that the instantaneous "inner" specific heat $=1.118+0.001176\,(t-32)$. The original data of the experiments are not given, but Professor Dieterici has kindly furnished the values which he actually obtained for the mean "inner" specific heat and also a copy of Mr. Drewes' dissertation containing the values obtained by him. If each of these values of the mean "inner" specific heat is multiplied by the temperature

range covered the result will be approximately equal to the internal energy u above 32 deg. fahr. at the temperature t. Since $i' = u' + Apv'$, in order to obtain values of i' the product Apv' corresponding to the temperature t must be added to the value of u' obtained as described. The results of these calculations appear in Table 6. Since Dieterici reported his results in terms of a mean calorie equal to 4.222 joules while in the present investigation a mean calorie of 4.184 joules is used, all of Dieterici's heat quantities must be multiplied by 1.009. The table includes the determinations made by Dieterici himself and all of the determinations made by Drewes; part of the latter are called reliable by Dieterici, while the rest are marked "falsch" in

TABLE 7 SUMMARY OF DETERMINATIONS OF THE SPECIFIC HEAT OF LIQUID AMMONIA

Temperature, Deg. Cent.	Temperature, Deg. Fahr.	Specific Heat of Liquid	Authority
62–30	144–86	1.22876	von Strombeck
20–0	68–32	1.02	Elleau and Ennis
77.7	139.9	0.886	Ludeking and Starr
20–16	68–61	1.094	A. J. Wood

the copy of Drewes' dissertation loaned by Dieterici. The ground for this statement is not given. In Fig. 8 Dieterici's values are plotted as black dots, and those of Drewes as circles. The values questioned by Dieterici are indicated by double circles. The curve represents the values found by the use of equations [14] and [23]. It is seen that in several cases the curve passes between the Dieterici points and the doubtful Drewes points.

73 Since the quotient obtained by dividing the change in i' between two temperatures by the difference in temperature is very nearly equal to the specific heat, the other determinations which have been made of the specific heat may be used to check the calculated values of i'. Table 7 contains a summary of the determinations of the specific heat of the liquid in addition to the work of Dieterici and Drewes already given; this includes the work of Elleau and Ennis, von Strombeck, Ludeking and Starr and A. J. Wood. In Fig. 9 these values are plotted, together with the curves representing the following equations used in the computation of various tables:

Dieterici	$c'=1.118+0.001156\ (t-32)$
Zeuner	$c'=1.01235+0.00468\ (t-32)$
Wood	$c'=1.12136+0.000438t$
Ledoux	$c'=1.0058+0.0020322\ (t-32)$
Elleau and Ennis	$c'=0.9834+0.0020322\ (t-32)$
Peabody	$c'=1.1$ constant

74 The full line curve represents the values of the tangents to the heat content curve of the present investigation. Since according

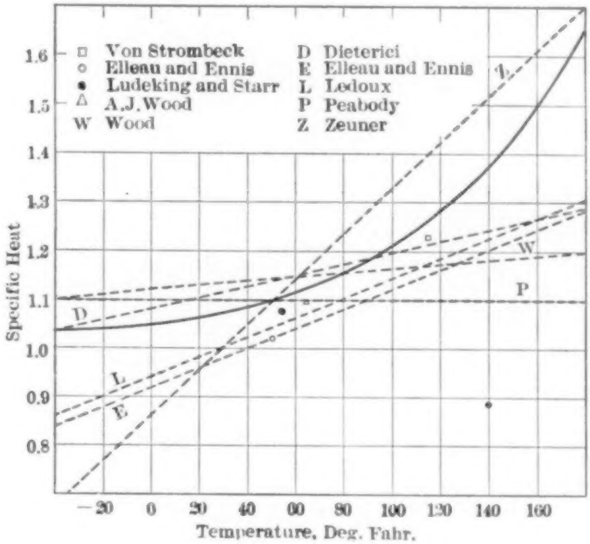

FIG. 9 COMPARISON OF VARIOUS DETERMINATIONS OF SPECIFIC HEAT OF LIQUID AND VARIOUS EQUATIONS PROPOSED TO REPRESENT SAME QUANTITY

to modern ideas of the critical point the specific heat of the liquid there becomes equal to plus infinity, the full line curve is of a more rational form than any linear relation.

75 *Entropy of the Saturated and Superheated Vapor.* An expression for the entropy is readily derived from the characteristic equation and the expression for c_p:

$$s=\alpha \log_e T+\beta T-AB \log_e p-Anp\frac{m}{T^{n+1}}+S_o \ldots\ldots\ldots [24]$$

For the complete derivation of this equation, see Appendix No. 3.

76 The constant s_0 is determined by passing to the saturation limit at 32 deg. fahr., where $s'=0$ and therefore $s''=\frac{r}{T}$. Substituting this value for s in the left-hand member of equation (s) and the proper values of p and T in the right-hand member, s_0 is found to be equal to -0.82656. Substituting this value and the known values of the other constants, and passing to common logarithms equation [24] becomes

$$s=0.8796 \log T+0.000174\ T-0.2695 \log p-\frac{p}{T^6}C-0.82656 \quad [25]$$

where $\log C=12.866554$.

ACKNOWLEDGMENT

77 Acknowledgment is hereby made to Prof. G. A. Goodenough of the University of Illinois, who suggested this investigation and who has given to it his hearty encouragement and his personal supervision.

78 The investigation was made under the auspices of the mechanical engineering department of the Engineering Experiment Station of the University of Illinois, and complete tables based upon the formulae set forth in this paper, together with a Mollier diagram and a full bibliography of the subject will in due time be published as a bulletin of the station.

APPENDIX No. 1

METHOD OF CONSTRUCTION OF FIG. 1

79 Using formula [7] to obtain the saturation temperatures of ammonia from those of water

$$\frac{1}{T_a}=c\frac{1}{T_w}+k$$

Let

$$\frac{-1}{T_w}=y \quad \text{and} \quad \frac{-1}{T_a}=x$$

then

$$-x=-cy+k$$

Since y is a function of the water vapor temperature corresponding to pressure p, y is also a function of p; the value of y for a given value of p may be found by making use of the tables of water vapor pressure; x is a function of the temperature of the ammonia corresponding to this same vapor pressure p; the curve giving x as a function of y is evidently a straight line.

80 Since y is a function of the pressure p and x a function of the corresponding temperature, the values of x and y may represent their respective values of pressure and temperature. The corresponding values of saturation temperature and vapor pressure may be read directly from the diagram.

81 Fig. 1 was constructed in the following manner: Integral values of temperature were assumed and the corresponding values of x were computed according to the relation $x=-\frac{1}{T}\cdot$ These x's were then laid off to a convenient scale and each labeled with the temperature to which the value of x corresponds. Integral values of pressure were assumed, the corresponding values of temperature found from steam tables, and the corresponding values of y computed according to the relation $y=-\frac{1}{T}$. These y's were then laid off to a convenient scale and each labeled with the pressure to which the value of y corresponds.

82 The value of vapor pressure at a given temperature as taken from the Marks and Davis steam tables was found to agree with the value given by the new Marks formula with sufficient accuracy up to a temperature of 400 deg. fahr.; above this temperature the values differed materially. In the construction of the chart, therefore, the steam tables were used to find the pressures corresponding to temperatures below 400 deg. fahr. and the newer formula was used to find the pressures corresponding to temperatures above that point.

83 Table 8 shows the calculations for a few of the values of Fig. 1.

84 In order to settle upon the most probable location of the straight line representing the pressure-temperature relation of saturated ammonia upon this

TABLE 8 SAMPLE CALCULATIONS OF THE COÖRDINATES OF FIG. 1

ABSCISSAE

Temperature, Deg. Fahr.	Absolute Temperature	Reciprocal of Absolute Temperature	Corresponding Value of X, In., referred to the point −10 as Origin $= -(8000 \times \pm -10)$
−100	359.64	0.0027806	12.244
−50	409.64	0.0024412	9.529
0	459.64	0.0021756	7.405
+50	509.64	0.0019622	5.697
100	559.64	0.0017869	4.295
150	609.64	0.0016403	3.122
200	659.64	0.0015160	2.128
250	709.64	0.0014092	1.273

ORDINATES

Pressure, Lb. per Sq. In.	Corresponding Saturation Temperature of Water Vapor, Deg. Fahr.	Absolute Temperature	Reciprocal of Absolute Temperature	Corresponding Value of y, In., referred to the point −18 as Origin $= -(20000 \times \pm -18)$
1	101.83	561.47	0.0017810	17.621
5	162.28	621.92	0.0016079	14.158
10	193.22	652.86	0.0015317	12.634
50	281.0	740.64	0.0013502	9.004
100	327.8	787.44	0.0012699	7.399
500	467.2	926.84	0.0010789	3.579
1000	544.9	1004.54	0.0009955	1.910
1700	613.4	1073.04	0.0009319	0.639

TABLE 9 SUMMARY OF DETERMINATIONS OF CRITICAL DATA FOR AMMONIA

Investigator	Date	Temperature, Deg. Cent.	Pressure, Atmospheres	Temperature, Deg. Fahr.	Pressure, Lb. per Sq. In.
Dewar.........	1884	130.0	115	266.0	1690.0
Vincent and Chappuis..	1886	131.0	113	267.8	166.08
Jaquerod.......	1908	132.3	109.6	270.1	1610.7
Sheffer.........	1910	132.1	111.3	269.8	1635.7

chart the experimental data must be plotted. The pressures observed by Pictet over a temperature range of −22 to 122 deg. fahr. agree almost exactly with those observed by Regnault; they are neither tabulated nor plotted. Faraday's results are quite inconsistent with those of Regnault, and are neither plotted in Fig. 1 nor given weight in the determination of the constants of the equation.

85 The two points in the upper right-hand corner of Fig. 1 represent the experiments at the critical point. The open square shows the values given by Vincent and Chappuis and the solid triangle the values given by Dewar. The values from which these points were plotted are given in Table 9.

86 This table also contains the values for critical data given by Jaquerod and Scheffer, which were found after the plate for Fig. 1 was made.

APPENDIX No. 2

DERIVATION OF EQUATION [19]

$$c_p = \phi T + \frac{Amn\ (n+1)}{T^{n+1}} p$$

87 The Clausius relation is expressed by the equation

$$\left[\frac{\partial c_p}{\partial p}\right]_T = -AT \left[\frac{\partial^2 v}{\partial T^2}\right]_p$$

The derivative in the right-hand member is determined from the characteristic equation. Thus from equation [18]

$$\frac{\partial v}{\partial T} = \frac{B}{p} + \frac{mn}{T^{n+1}} \qquad [26]$$

and

$$\frac{\partial^2 v}{\partial T^2} = -\frac{mn\ (n+1)}{T^{n+2}} \qquad [27]$$

Substituting the second derivative in the Clausius relation the result is

$$\left[\frac{\partial c_p}{\partial p}\right]_T = \frac{Amn\ (n+1)}{T^{n+1}} \qquad [28]$$

Taking T as a constant and integrating with respect to p as the independent variable, the result is

$$c_p = \frac{Amn\ (n+1)}{T^{n+1}} p + \text{constant of integration}$$

The constant of integration may be a function of T, since T was held constant during the integration; hence

$$c_p = \phi\ (T) + \frac{Amn\ (n+1)}{T^{n+1}} p \qquad [29]$$

APPENDIX No. 3

DERIVATION OF EQUATIONS [22] AND [24]

$$i=\alpha T+\frac{\beta T^2}{2}-A\ (n+1)\ p\frac{m}{T^n}-Ac_p+i_o$$

$$s=\alpha\ \log_e T+\beta T-AB\ \log_e p-Anp\ \frac{m}{T^{n+1}}+s_c$$

88 Having an explicit formula for the specific heat at constant pressure in terms of the variables p and T, an expression for the heat content in terms of these variables may be easily derived. For this purpose the general equation

$$dq=c_p dT-AT\left[\frac{\partial v}{\partial T}\right]_p dp\ldots\ldots\ldots\ldots\ldots\ldots\ldots[30]$$

is most convenient. Since the heat content is defined by the relation

$$i=A\ (u+pv)$$

then

$$di=A\ [du+d(pv)]$$

or

$$di=dq+Avdp$$

Hence by substitution in equation [30] there results

$$di=c_p dT-A\left[T\frac{\partial v}{\partial T}-v\right]dp\ldots\ldots\ldots\ldots\ldots\ldots\ldots[31]$$

From the characteristic equation

$$\frac{\partial v}{\partial T}=\frac{B}{p}+\frac{nm}{T^{n+1}}$$

whence

$$T\frac{\partial v}{\partial T}-v=(n+1)\ \frac{m}{T^n}$$

Substituting this and the general expression for c_p in equation [31] the result is

$$di=(\alpha+\beta T)\ dT+Amn\ (n+1)\ p\frac{dT}{T^{n+1}}-\frac{Am\ (n+1)}{T^n}\ dp-Acdp\ldots\ldots.[32]$$

Since i depends upon the state of the substance only, the second member of equation [30] must be an exact differential. The integral is readily found to be

$$i = \alpha T + \frac{\beta}{2} T^2 - A(n+1)\frac{pm}{T^n} - Acp + i_o \qquad [33]$$

89 An expression for entropy is readily found from equation [30]. Thus

$$ds = \frac{dq}{T} = c_p \frac{dT}{T} - A\left[\frac{\partial v}{\partial T}\right]_p dp \qquad [34]$$

Introducing in equation [34] the expressions previously derived for c_p and $\left[\frac{\partial v}{\partial T}\right]_p$, the result is

$$ds = \left[\frac{\alpha}{T} + \beta\right] dT + Amn\ (n+1) p \frac{dT}{T^{n+2}} - \frac{ABdp}{p} - \frac{Amn}{T^{n+1}} dp \qquad [35]$$

The integral of equation [35] is

$$s = \alpha \log_e T + \beta T - AB \log_e p - Anp \frac{m}{T^{n+1}} + s_o \qquad [36]$$

APPENDIX No. 4

TEST OF VALIDITY OF EQUATION [23] BY MEANS OF THE LAW OF CORRESPONDING STATES

90 Since i is constant in a throttling process, the Joule-Thompson coefficient μ may be defined as the derivative

$$\left[\frac{\partial T}{\partial p}\right]_i$$

From calculus

$$\left[\frac{\partial T}{\partial p}\right]_i = -\frac{\frac{\partial i}{\partial p}}{\frac{\partial i}{\partial T}}$$

and from the definition of heat content i

$$\frac{\partial i}{\partial T} = c_p$$

Hence

$$\mu = \left[\frac{\partial T}{\partial p}\right]_i = -\frac{1}{c_p} \cdot \frac{i}{p}$$

or

$$\mu = \frac{A}{c_p}\left[\frac{m\ (n+1)}{T^n} + c\right] \qquad [37]$$

91 From equation [37] it is seen that μ varies with the pressure; as the temperature rises, however, the influence of pressure decreases. Joule and Kelvin, working with gases far removed from the saturation limit, found that μ varies inversely as the square of the absolute temperature. The experimental values of μ for steam expressed in pounds per square inch and degrees fahrenheit were reduced by multiplying by 2.56, a factor which is the ratio of the critical pressure of water (2947 lb. per sq. in.) to the critical temperature (1149 deg. fahr.), these critical values being the ones determined by Cailletet and Colardeau. Davis gives a curve which he says represents the experimental values for water in the best possible manner; the values in this curve are those translated back to ordinary units. Later determination of the critical constants for water by

Holborn and Bauman give 3200 lb. per sq. in. for the critical pressure and 1166 deg. fahr. for the critical temperature Table 10. If then the values of μ on the

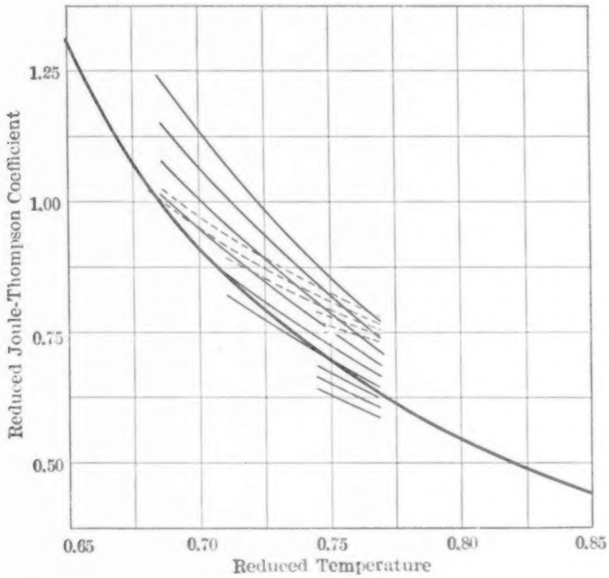

FIG. 10 COMPARISON OF VALUES OF REDUCED JOULE–THOMPSON COEFFICIENT FOR AMMONIA ACCORDING TO WOBSA AND VALUES RESULTING FROM EQUATION (p) WITH VALUES FOR WATER AS GIVEN BY DAVIS

TABLE 10 SUMMARY OF DETERMINATIONS OF CRITICAL DATA FOR WATER

Investigator	Date	Critical Temperature, Deg. Fahr.	Critical Pressure, Lb. per Sq. In.
Nadejdine	1885	676.6	.....
Battelli	1890	687.7	2859
Cailletet and Colardeau	1891	689.0	2994
Strauss	1892	698.0	2873
Traube and Teichner	1904	705.2	....
Holborn and Bauman	1910	706.3	3200

Davis curve are multiplied by $\frac{3200}{1166}$, or 2.745, and the temperatures are divided by 1166, there results a curve expressing the variation of μ with temperature, both expressed in reduced units, according to the best available experimental data.

TRAIN LIGHTING

By H. A. Currie and Benjamin F. Wood

ABSTRACT OF PAPER

This paper reviews the subject of train lighting from the time candles were used to the present day. The methods employed are given in historical order and comprise the candle, oil, gas, and electric lighting systems. The systems appear to occupy a period of 25 years each, beginning with candle lighting in 1825. The early methods of candle and oil lighting are described in brief outline. The gas lighting period is gone into more fully and the systems in general use are described. The electric lighting period is naturally discussed in still greater detail.

The development of the present systems from the time primary and storage batteries were first used is gone into, and the present day straight storage, head-end, and axle equipments are described in detail. Under axle generating systems, the various requirements for successful operation are discussed and the various types of suspension are described. Questions of suitable voltages, lamps, deflectors, and fixtures are taken up and standardization of voltages and batteries are discussed.

Tabulation showing the number of electrically lighted cars, cars to be electrically lighted, types of battery, voltages, number of cells, etc., is included, as is also a train lighting bibliography arranged in chronological order.

TRAIN LIGHTING

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The first railway trains, usually made up of one car, made short trips in the day time only and artificial light was not required; as the trips were extended into the evening the cars were still unlighted. The lighting of stage coaches before the day of railway coaches was obtained by candles brought in by the passengers themselves and placed in sockets along the sides of the coach. It is recorded that the first railway car illumination can be traced to one Thomas Dixon, a driver on a horse train railway in England in 1825, who out of his generosity placed a penny candle on a table in the center of the car for the convenience of his passengers. The early railroad patrons who contemplated traveling by night, and were foresighted, brought their own candles; and later the candles were supplied by the railroad companies in crude sockets, but the passenger had to light them and nurse the flame.

2 A review of the development of railway car lighting shows that it is divided into periods of approximately twenty-five years for each of the illuminants used, beginning with candles in 1825. In 1850 oil lamps were introduced, which gave way to gas in 1875, and this in turn was superseded by electricity in 1900. The development of each of these systems extended over a number of years and the new illuminant did not entirely displace the previous method of lighting. In fact candles are

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sometimes used today for emergency lighting, while oil lamps may be found in many coaches. With the early applications of electric lighting to cars it was usual to retain gas lights for emergency use. Increased reliability of the electric system has now made this precaution unnecessary. Light failures on well-maintained electric service now average in the neighborhood of 1 per 1,000,000 miles.

CANDLE PERIOD 1825 TO 1850.

3 The railroad companies first developed the candle light by placing the candles in proper sockets, putting chimneys around them and deflectors on top of the chimneys. Later the candle flame was kept at a predetermined focus by a spring in the candle socket. The candles used in this country were made from the solid distillates of petroleum formed in molds under pressure, while in Europe wax and tallow dips were chiefly used.

OIL PERIOD 1850 TO 1875

4 The railroad companies recognizing the need of better illumination tried various forms of oil lamps, and in this development the lamps were always the latest improvement as regards safety and reliability. The oils used in this country were distillates of petroleum differing in flame and fire test in the various states and on different roads. The oils used in Europe were vegetable oils, because of the high cost of importing mineral oils and because of the supposed danger from fire. The danger of fire from oil of either class depended, however, more upon the method of containing than in the kindling point, except in the very low flash point distillates of petroleum which were not used directly in railroad work.

5 The European practice for oil lamps was to use rape seed and Colza vegetable oils which are irregular in composition and congeal in cold weather. The difficulties met with in the burning of this oil in Europe led to the development of central draft burners, which were also used in this country to burn heavy oils. The early types of mineral oil burners were merely reservoirs into which fibrous wicks were dipped through suitable holders. Two wicks feeding one flame was the next development, allowing a more copious supply of oil to the flame and a brighter light resulted.

6 The Argand and Belgian burners are representative types of the central draft burner. The Argand burner has a tube and annular wick providing a strong draft of air both to the inside and to the outside of the flame. This results in a high temperature and a bright white light. This burner was the forerunner of several central draft burners used for car, locomotive and station lighting, where a high candle power was desired. A maximum of 60 c.p. has been obtained from one lamp of a size suitable for railway cars by this central draft method. The Belgian lamp represents the highest improvement in delivering a current of air both inside and outside the flame so that it will impinge on the flame at the proper point to produce rapid combustion. Other burners of this type are the Acme, the Student Lamp, and the Westlake lamp.

7 The reservoirs of these lamps were generally of metal and either double glass or specially tough single glass chimneys were used. When the lamps were placed in the center of the ceiling of the car, smoke bells were provided to take off the products of combustion and to protect the flame from drafts. The oil lamps had been generally superseded for first-class service by other illuminants before the development of the incandescent mantle, and therefore, has not been used with the oil lamp in train lighting.

GAS PERIOD 1875 TO 1900

8 About 1870 coal gas was introduced as a head-end system on several railroads as a means of lighting cars. A reservoir in one of the cars, known as the guards van, was filled with gas secured from the city mains and then slightly compressed by water displacement. The reservoir which had a capacity of about 100 cu. ft. was generally square in shape and was made of strong canvas reinforced by wood ribs. Gas was conveyed to the lamps, which burned about 5 cu. ft. each per hour, through iron pipes with rubber hose connections between cars.

9 Nearly ten years later Mr. George Westinghouse devised the "air gas" or carbureted air system. A heavy cylinder, suspended underneath each car, was filled with such material as sawdust, felt, etc. saturated with some light hydrocarbon such as benzine. Compressed air was passed through the tank and, becoming saturated with the hydrocarbon vapor, was conveyed to the lamps where it was burned. The system was never ad-

vanced much beyond the experimental stage, as changes in temperature affected the carburation to a great extent.

10 Carbureted gasolene, made by mixing air and gasolene in suitable proportions was the next system to be used. The best example is probably the Frost Dry Carbureter System. The essentials of this system are that the gasolene be kept from leaking and that the gas be burned as soon after generation as possible without undergoing any change of temperature. The Frost system accomplishes this by locating on the roof of the car directly over each lamp, the carbureter which consists of a chamber filled with wicking soaked in gasolene through which air from the air-brake system is passed at a low pressure, the gasolene vapor being supplied to a burner directly underneath. The products of combustion are passed out through a smoke bell which leads around the carbureter, thus tending to keep the gas at the proper temperature in all weather. The difficulty of maintaining this proper temperature, however, was the chief drawback of the system.

11 The extensive manufacture of calcium carbide by the electrochemical process gave the acetylene system a great impetus. Examples of the acetylene system in extensive use were first, the Adlake system in which about 50 lb. of calcium carbide, placed in a portable steel container installed in a special panel in the car, is arranged so that gas is generated automatically as required by water dropping upon the carbide. In order to obtain this automatic supply, a small tank is placed underneath the car and connected to the steel container, which automatically produces the gas because of the back pressure of the water supply when a certain amount of gas has been generated.

12 The Avery system resembles the Adlake system, except that the generator is placed underneath the car body.

13 The third system used to some extent, is known as the Commercial and in this, acetone, a liquid distilled from wood alcohol and having the property of absorbing acetylene under pressure, is used. The gas is stored in tanks filled with asbestos discs saturated with acetone, and is given off as the pressure in the tank is reduced.

14 Acetylene gas gives a more intense light than any of the other oil or gas systems, but it is explosive and this fact has militated against its extensive use.

15 In 1867 Pintsch, in Berlin, devised the system of gas illumination which bears his name. The distinctive features of this system are, the manufacture of gas from crude petroleum by heat distillation, which gas is not decomposed by compression and subsequent expansion and a regulator which controls the supply of gas to the lamps at a very low pressure for atmospheric burning. Because of its reliability and increased storage capacity, obtained by charging the reservoirs at a high pressure, the Pintsch gas is very extensively used for car lighting. Other systems previously tried, namely, coal, water, and natural gas, did not have these distinctive features and up to that time, were not a success. The substitution of mantle burners for the ordinary fish-tail burners has greatly improved the quality of light, and enabled the Pintsch system to compete on more equal terms with electric lighting.

16. The Pintsch system requires a separate plant with appliances for generating, compressing and storing the gas, also suitable tank cars and pipe lines for conveying the gas to the car reservoirs.

ELECTRIC PERIOD 1900 TO DATE

17 Early in 1881 the London, Brighton and South Coast railway had in operation a straight storage battery system with a battery under each car, the battery being charged during the layover period at terminals.

18 Bichromate primary batteries were tried on trains in France during the year 1885. The scheme was abandoned, however, when the first set of batteries had run down and renewal of the electrodes and acid was necessary.

19 About this time both the Pennsylvania Railroad and the Boston and Albany tried out the straight storage system. The general opinion was that the system was practical but too expensive for general use.

20 In 1888 the Chicago, Milwaukee and St. Paul had in service a train equipped with a head-end system consisting of a boiler, engine, generator, etc., installed in a separate car for supplying light to the train. The present system, developed later, uses steam from the locomotive.

21 The axle driven generator was developed coincidently with electric lighting. The first installation of this kind consisted of a generator connected to a relatively high capacity

battery and hand-operated main switch and pole changer, regulation being effected through the inherent regulation of the batteries taking the excess current at high speeds, thereby holding the voltage nearly constant by armature reaction. Later the brushes were automatically shifted with the speed for voltage regulation. The next step was to allow the belt to slip at speeds in excess of the speed required to deliver current at normal voltage.

22 Various early and intermediate systems were proposed, among which may be mentioned:

a A head-end system deriving power from a fan-driven dynamo on the pilot.

b A track wire system, especially through tunnels, for lighting cars from a trolley.

c A track wire system for charging batteries from an auxiliary trolley while cars were passing through electrified territory.

23 The working out of the early tried system resulted in the more or less general adoption of the following: (*a*) the straight storage system, (*b*) the head-end system, (*c*) the axle generator system.

STRAIGHT STORAGE SYSTEM

24 In this system, each car is provided with a storage battery, which must be charged at terminals during the layover period. A consideration of the requirements for successful operation reveals the following essentials:

The capacity of the battery must be in excess of the demand for current to operate lamps, fans, etc., for the longest run between charging periods.

The power plant, or other outside source of power, must be of sufficient capacity to meet the maximum demand for charging current.

The layover time at terminals must be sufficient to cover all necessary shifting and charging of the batteries at the proper rate.

The yard must have a sufficient number of tracks provided with charging outlets, so arranged that the charging of batteries will not interfere with shifting operations.

25 Two considerations enter into the determination of the

size of the battery; the demand for current and the weight of the battery. The demand for current will vary with the type of car used, the number and size of lamps and other electrical apparatus installed and the time during which this apparatus is used. The battery must be divided into units of a weight that may be readily handled.

26 The batteries in general use in train lighting service have a rated capacity of approximately 300 ampere-hours. This

TABLE 1 SHOWING COSTS OF VARIOUS SYSTEMS OF ILLUMINATION PER CAR

	1825–1850 Candles	1850–1875 Oil	1875–1900 Gas			1900 Electric		
			Gasolene	Pintsch	Acetylene	Head-End	Straight Storage	Axle Coach
Passengers carried	4–20	20–50	50–60	60–70	60–70[1]	30–40	[1]40–50	60–70
Candle Power	4–10	20–60	200	300	400	350	250	450
Fixtures and Connections	0 to $3	$10–$60	$500	$400	$500	$700[3]	$600	$1000
[3] Cost of Plant	.....	.....	...	60[5]	$40	25	20	....
[2] Yearly Operating Cost	15–25	40–120	150	150	$200	750	500	200
Yearly Haulage Cost	.....	.....	150	200	$200	400	300	400
Interest, Insurance and Taxes	.....	.....	50	50	$60	100	62	145
[4] Total Yearly Cost	20	60–100	350	400	$500	1250	1350	750
Cost per Year per Seat	1	1.50–2	6	7	$8	31	34	12

[1] At the time of the prevalence of these systems only limited trains were so equipped.
[2] Including all maintenance.
[3] Share per car considering whole train.
[4] Average
[5] Based on capacity for 500 cars.

Basis for above figures, Passengers Carried suppose all seats to be occupied. The Fixtures and Connections are in nearly every case from actual installations. Cost of Plant is the first cost divided by the capacity in number of cars supplied. Yearly Operating Cost includes fuel or power and attendance and maintenance but not all are actual costs, but are filled out by close estimates. Yearly Cost per Seat is simply an arbitrary means of comparing cost per passenger.

Table 2 Methods and Distributions of Car Lighting Systems.

is about the maximum limit of capacity for Plante's type batteries having weight low enough for convenience in handling.

27 The great majority of cars on which this system is used are equipped with a 64 volt lead-lead, or nickel-iron battery with cells connected in series. A few cars are operated at other voltages, viz. 26, 30, 32 and 110. It was the practice of one railroad some years ago to operate at 25 volts, using 12 lead-lead cells in series, with two or more 12 cell sets on a car connected in multiple.

28 Two battery boxes are generally provided and secured

to the under side of the car, one on each side equidistant from the ends of the car, and with the front or door side slightly back of the line of outside finish. The cells are put up in double compartment lead-lined wood tanks provided with handles, rollers, etc., for convenience in handling.

29 The two halves of the battery are connected in series and leads are run to the switchboard in the end of the car. Taps are taken off these leads at the battery terminals and run to charging receptacles, conveniently located on each side of the car.

30 The ampere-hour meter is coming into general favor as an indicator of the state of charge of the battery. As an adjunct a shunt trip circuit breaker is sometimes installed, the connections being such that, when the battery is fully charged, the pointer on the meter closes a circuit which energizes the shunt trip and opens the breaker, thus cutting the battery off charge.

31 The source of power available for charging purposes at terminal yards must be of the proper voltage and of capacity sufficient to meet the maximum demand. The charging voltage provided is usually 50 per cent higher than the normal voltage of the battery. Hand or automatically operated resistance devices are provided for reducing the voltage of the individual charging lines to the proper point.

32 The batteries are charged while on the car under normal conditions. When the layover period is short it is sometimes necessary to exchange a discharged battery for one fully charged.

33 The operating schedule of cars equipped with this system must be worked out to allow sufficient layover at terminals to permit the charging of batteries and shifting of cars. Anything that restricts the shifting operations necessitates either additional yard trackage, additional motive power, or both. As cars produce a revenue only when in service, it is the aim of the transportation department to decrease the layover period to a minimum.

HEAD-END SYSTEM

34 The head-end system consists essentially of a steam-driven generator located in the baggage car or on the locomotive. Proper controlling apparatus is provided and train lines are run from the generator through the entire length of the train, flexible connections being used between cars.

35 So far as can be learned the first head-end system was installed by the Pullman Palace Car Company in January 1887, in some of their composite cars. A three-cylinder Brotherhood engine, direct-connected to an Eickemeyer generator, rated at 80 amperes and 80 volts full load at 1200 r.p.m. was used. These cars were operated in the Atlantic Coast Line Special running between Jersey City, N. J., and St. Augustine, Fla. In June of the same year the Pennsylvania Limited was equipped with the same type of apparatus. In both of these trains the individual cars were equipped with batteries.

36 This system of lighting was afterwards taken up by other eastern railroads but at the present time it is used chiefly on roads operating west of Chicago. Following the Brotherhood engine, the Westinghouse standard engine, the De Laval turbine and the Curtis turbine entered the field in turn.

37 The Pennsylvania Railroad conducted a series of experiments with the "Gould Booster" head-end system, which consisted of a compound-wound turbine-driven generator mounted on the locomotive, with a motor-generator booster and automatic switches mounted on the tender. Four train line wires were run underneath the cars and were joined together by connectors that uncoupled automatically when the cars were parted. A battery and transfer switch were provided on each car. In this system, unlike any other in use at that time, the entire control was automatic after the generator was started.

38 The head-end system of today comprises the following apparatus:

A generator, usually steam turbine-driven, placed in the baggage car or on the locomotive, and furnished with steam from the locomotive.

The necessary indicating, regulating, and controlling apparatus placed near the generator and in an accessible position.

Train line wires of the proper size on each car and running the entire length of the train, flexible connections being made between cars, in the vestibule.

Batteries, consisting of a suitable number of cells connected in series and placed in battery boxes attached to the under side of the cars.

Lamp regulators are sometimes installed in the cars to

compensate for the line drop and to maintain constant voltage at the lamps.

39 The successful operation of this system requires that a sufficient amount of steam at the proper pressure be provided when lighting is necessary. As it is the object of the transportation department to get trains to their destination on time, lack of steam is felt first by the lighting system, the pressure being reduced or steam cut off entirely so that the schedule may be maintained.

40 When the train is broken-up en route, it is obvious that each section must either be equipped with a battery to insure light until the train is again made-up, or provided with some auxiliary light.

41 A member of the train crew must be capable of operating the generating apparatus and of making running repairs and adjustments en route.

42 Head-end systems are generally operated at 64 or 110 volts, although in late years the introduction of tungsten lamps has gone a long way to eliminate the need of the high voltage equipments and comparatively few railroads are now using 110 volts.

43 It is apparent that the use of the head-end system must be restricted to a few trains having assigned runs or that it must be extended to cover all the cars operated by the railroad in electrically-lighted trains.

AXLE GENERATOR SYSTEM

44 The axle generator systems used in this country comprise the following principal parts:

- *a* An axle-driven generator mounted on the car truck. (Abroad where rigid trucks are used the axle generator is frequently secured to the under side of the car body.)
- *b* A suspension by which the axle generator is supported from the truck frame.
- *c* A drive, connecting the armature shaft to the axle.
- *d* A regulator for controlling the voltage and output of the generator at all train speeds.
- *e* An automatic switch designed to open on reverse current for the purpose of preventing discharge of the battery through the generator.

f A regulator for controlling the voltage impressed on the lamp circuits.

g A battery of a suitable number of cells to supply current when generator current is not available.

45 For the successful operation of the system, the following requirements must be met:

The polarity of the generator terminals must remain unchanged with a movement of the car in either direction.

At all train speeds, from the cutting-in speed of the generator to the maximum, the generator output and voltage must be maintained within the desired working limits.

The generator must be automatically connected and disconnected from the battery circuit as the train speed rises above or falls below the critical speed.

The lights may be burned at any time and the transfer of this load from the battery to the generator and vice-versa must result in no appreciable change in the candle power of the lamps.

The voltage impressed on the lamp circuit must be maintained within such limits as will give satisfactory illumination and reasonable life of lamps.

46 The early axle generators were practically all of the constant current type. The generator maintained a constant current, only that portion going to the battery that was not required by the lights. The generator output could be varied by hand adjustment, and it was necessary to adjust the generator output so that there would be an approximate balance between the battery input and battery output, account being taken of the battery ampere-hour efficiency. This requires consideration of the following factors: (*a*) the ampere capacity of the generator, (*b*) the ampere-hour demand during the trip. The demand varies with the number and size of lamps installed, the run in which the car is operated, the number and duration of stops, and the proportion of total running time of the train below cutting-in speed. A change of the car from local to through service, or vice-versa, required a complete readjustment of the regulation. A failure to so readjust would quickly result in battery deterioration, the plates becoming sulphated with a low generator output, while a corresponding high output would

boil away the electrolyte. In a variation of this method the battery was charged at a definite rate, generally the normal charging rate, while the generator output varied with the lamp load. In more recent systems constant current is maintained until the generator voltage, increasing as the counter e.m.f. of the battery, reached a definite value, after which one of two results would obtain, depending upon the design of the control apparatus. In one the generator voltage was held approximately constant producing what was known as the taper charge or a gradual decrease in the rate of charge. In the other, the generator voltage was automatically reduced to the "floating" e.m.f. of the battery and maintained approximately constant. This was known as "stop charge" regulation.

47 Both methods originally depended upon the operation of a voltage coil actuating a switch, which in turn changed the field current by means of resistance. Premature closing of the battery stop charge switch resulted in undercharging and consequent sulphating, while delayed closing caused overcharging and the boiling away of the electrolyte.

48 These devices are being superseded by a system using a voltage coil in the regulator instead of the voltage switches previously mentioned. Here the current output is maintained constant to the full generator capacity regardless of the current demand, and the battery must supply all current in excess of the capacity of the generator. As the battery becomes charged the generator voltage gradually increases and at a predetermined voltage the voltage regulating coil takes the control of the generator from the current regulating coil and when the battery becomes fully charged the current decreases until the battery is "floating on the line." As the voltage on the battery on charge is approximately 30 per cent higher than on discharge, it is necessary to provide some means of lamp regulation in order to keep a constant voltage on the lamps. In the early axle generating systems this was accomplished by introducing a fixed metallic resistance into the lamp circuit at the instant the automatic switch closed transferring lamp load from battery to generator. This method, however, was not altogether reliable as the generator voltage varied with the condition of the battery and the number of lamps burning. The variation of the voltage due to these conditions soon led to the development of a variable metallic resist-

ance in place of the "fixed" resistance. In this method of regulation the amount of resistance in the circuit was varied by means of the operation of a small auxiliary motor operated by a voltage relay or other means.

49 A later type of lamp regulator employed, as a means of regulating resistance, a series of carbon blocks, the resistance being varied by varying the pressure on these blocks, the variation of pressure being determined by a pilot voltage coil connected across the lamp mains.

50 In this country it is the general practice to support the axle generator from the truck frame. When first applied, the generator was placed between the axle and the truck-end sill, this arrangement being known as "inside suspension." The generator was not easily accessible for inspection and repairs, and at the present time it is placed outside of the truck frame, this arrangement being known as "outside suspension." There are four general methods of carrying the generator from the suspension framing, viz. bottom pivoted, top pivoted, parallel link and sliding. The bottom pivoted was first used but at the present time the parallel link suspension is in more general use.

51 The drive usually used employed a rubber-filled canvas belt running on pulleys on the axle and the armature shaft. The axle pulley as first used was cast iron mounted directly on the car axle, the bore of the pulley conforming to the taper of the axle, but on account of inequalities in the axle which was hammered or rough-turned, it was wrapped with tarred paper. The axle pulley at present in use is of pressed steel, mounted on a steel bushing, the bushing being secured independently to a turned seat on the axle, and the pulley mounted thereon. Belt tension is provided by means of springs which also afford relief to the belt due to the movement of the car axle with respect to the truck frame. One spring is generally used when the generator has top, bottom or sliding suspension and two springs with the parallel link suspension. Chains of the silent type have also been tried and have the advantage of positive action and decrease in bearing pressure, but the wear of the links both on the face and the pivot sprockets has been excessive. Belts of V section have been tried and would seem to have the same advantages as the chains, but it is found that in winter the bottom of the V groove in the sheaves packs with

ice and snow, and driving power is lost. Neither the chain nor V belt requires tension device.

52 A form of shaft drive, from a bevel gear on the car axle, through an extensible shaft with universal couplings to a generator carried from the car body, is being tried. A gear drive, with the generator mounted on the car axle after the manner of the street car motor mounting has also been prepared and will soon be tried. The latter will no doubt require a track pit for the inspection and repair of the generator.

53 Plain bearings with ring oilers were used almost exclusively until five years ago. With this method of lubrication, it was necessary to carry the oil level so high that it frequently entered the generator frame and damaged the armature and field coils. To overcome this trouble, a form of wick oiler was tried which, however, proved unsatisfactory. The next improvement was a combination ring and chain oiler, which is now in general use on one type of machine. Waste-packed bearings have also been extensively used and have given no trouble on account of oil entering the generator frame. Considerable trouble, due to hot bearings has been experienced. Ball bearings for axle generators were introduced in England about 1907 and tried in this country in the early part of 1911. Although the use of ball bearings to date has been limited, the indications are that this bearing will become very popular on future machines. On account of the widely varying temperatures between summer and winter conditions that obtain in the operation of axle generators, it was necessary to develop special oils that would remain fluid at low temperatures. Those now used have a freezing point of 10 deg. fahr. For ball bearing lubrication grease should be free from acid or alkali; should not oxidize or evaporate; should not gum or lose its body.

LAMPS AND VOLTAGE

54 The early train lighting lamps were of the ordinary multiple burning type regularly manufactured, and as the demand increased manufacturers took up the development of lamps especially adapted to train lighting. One of the requisites of these lamps is that the smallest size bulb consistent with reasonable life of lamp be used.

55 Among the early lamps was a foreign-made carbon-filament lamp operating at 4 watts per c. p. Some time later, a

carbon lamp operating at 3.5 watts per c. p. was developed by American manufacturers.

56 It was found that the future of electric car lighting depended to a large extent upon whether or not lamps of still higher efficiencies could be manufactured in large quantities. About this time the tungsten lamp was introduced in regular multiple burning service and investigations were immediately started with a view to adapting it to car lighting. The use of tungsten lamps, however, was looked upon with some doubt on account of the extremely fragile nature of the filament, especially in the smaller sizes. Eventually a 25-watt lamp was developed. This did not improve conditions appreciably, since many of the carbon-filament lamps were approximately of the same wattage. Further investigations resulted in the development of a 15-watt lamp which improved conditions sufficiently to warrant its use in large quantities and made it possible to light a car at a reasonable current demand. With the introduction of tungsten lamps the standard voltage of axle generators has been changed from 60 to 30 volts.

57 Coincident with the introduction of tungsten lamps into train lighting service there was introduced the so-called "hot filament" system. The basis for this system was the belief that a tungsten filament of the early type when heated to about 400 deg. fahr., was less fragile than when cold, so that by maintaining the filaments at this temperature the life of lamps could be materially increased. A circuit switching arrangement was provided whereby the lamps were operated across 30 cells of battery when light was desired and across the two remaining cells of the battery at other times, this being sufficient to keep the filaments at a dull red. Later, the introduction of the drawn wire filament rendered this system unnecessary, and it has been abandoned.

58 Metalized-filament lamps with small opal bulbs, operating at about 2½ watts per c. p. were developed for berth lighting. These lamps gave satisfaction at first, principally because berth lights had never before been furnished. As the public became educated to their use, the demand for more light in berths became pronounced, so that it has been found necessary to use the 15-watt tungsten lamp as a berth lamp, and the use of metalized filament berth lamps is decreasing very rapidly.

59 Tungsten lamps for train lighting purposes are now furnished in 10, 15, 20, 25 and 50-watt sizes, although the bulk of

TABLE 2 METHODS AND DISTRIBUTION OF CAR-LIGHTING SYSTEMS

Railroads	Cars Lighted by Electricity	Cars Lighted by Other Means	Straight Storage		Cars with Turbine	Head-End System		Axle Generator System		Owned and Operated by Railroad Company	Owned and Operated by Pullman Company	Cars Contracted for	Number Cells Lead Battery	Number Cells Nickel, Iron, Alkaline Battery
			No.	Volts		No.	Volts	No.	Volts					
A., T. & S. F.	682	1120	0	0	0	0	0	682	30	632	50	50	11552	25
B. & A.	7	434	0	0	0	0	0	0	0	7	0	0	708	0
B. & O.	246	1105	28	60	11	67	64	19	60	143	103	0	3028	100
	..	..	..	..	..	..	..	29	30	..	..	..	..	..
Can. Pac.	165	2273	0	0	0	0	0	128	24	128	0	37	3072	220
C., St. P., M. & O.	58	265	0	0	4	0	64	2	60	2	0	0	192	0
C. of Ga.	6	258	0	0	0	0	0	6	30	0	0	0	80	50
C. & A.	103	100	56	32/64	0	0	0	34	32/64	103	0	0	0	0
C. & E. I.	93	107	2	32	0	0	0	77	32	79	14	0	1264	0
C., B. & Q.	710	591	0	0	52 cr	52	60	2	30	0	0	0	5508	0
	..	..	..	..	16 en	16	64	..	..	..	..	..	..	..
C., M. & St. P.	990	368	30	64	58	58	64	9	64	930	..	60–64	3872	0
	..	..	..	..	..	..	..	2	32	..	..	..	..	..
C. G. W.	70	102	0	0	0	0	0	70	30	70	0	0	736	216
C. & N. W.	672	1000	0	0	23	200	64	17	60	572	100	0	2200	25
	..	..	..	..	..	..	..	35	30	..	..	..	..	..
C., R. I. & P.	344	805	0	0	0	0	0	301	30	0	0	36–30	5024	125
	..	..	..	..	..	..	..	6	60	..	..	1–60	..	..
C., C., C. & St. L.	46	552	2	0	0	0	0	44	32/64	103	0	0	0	0
D., L. & W.	44	738	0	0	0	0	0	44	30	0	30	0	704	0
	..	..	..	..	..	..	..	1	60	..	..	..	..	..
D. & R. G.	30	348	0	0	0	0	0	29	32	0	0	0	512	0
E. P. & S.W.	17	46	0	0	0	0	0	17	30	0	0	0	272	0
Erie	156	1788	74	30	0	0	0	72	30	146	0	10–30	2448	0
G. N.	551	402	11	110	45	550	110	1	30	953	0	0	3040	0
G. R. & I.	33	86	28	60	0	0	0	1	60	33	0	0	912	0
	..	..	..	..	..	..	..	4	30	..	..	..	..	..
G. T.	106	0	0	0	0	0	0	15	24	31	75	0	840	0
	..	..	..	..	..	..	..	15	30	..	..	..	..	..
	..	..	..	..	..	..	..	1	60	..	..	..	..	..
F. W. & D. C.	22	31	0	0	0	0	0	22	32	0	0	0	336	25
Ill. Cen.	330	770	10	30	0	0	0	320	30	240	90	0	2240	3050
	..	..	..	..	..	..	..	3	60	..	..	..	..	..
K. C. So.	21	57	0	0	0	0	0	18	30	0	0	0	384	0
L. V.	219	340	0	0	0	0	0	196	32	0	0	23–30	3480	24
L. I.	113	404	3	64	0	0	0	109	32	0	0	0	3680	125
L. S. & M. S.	139	634	0	0	0	0	0	72	30	0	0	0	0	0
	..	..	1	64	..	..	..	47	60	..	..	..	3312	..
L. & N.	67	558	1	32	0	0	0	59	32	0	0	532	1120	75
M. P.	208	739	1	32	0	0	0	109	32	0	178	0	1824	0
M. & O.	6	119	0	0	0	0	0	6	..	6	0	0	0	0

TABLE 2—CONTINUED

Railroads	Cars Lighted by Electricity	Cars Lighted by Other Means	Straight Storage		Cars with Turbine	Head-End System		Axle Generator System		Owned and Operated by Railroad Company	Owned and Operated by Pullman Company	Cars Contracted for	Number Cells Lead Battery	Number Cells Nickel, Iron, Alkaline Battery
			No.	Volts		No.	Volts	No.	Volts					
M. & St. L.	19	76	0	0	0	0	0	19	32	17	2	0	304	0
M. C.	60	438	0	0	0	0	0	18	60	0	0	12–30	1248	0
	..	..	..	..	..	..	..	30	30	..	..	..	..	..
M., St. P. & S. St. M.	65	328	0	0	0	0	0	65	32	0	20	0	1072	0
N.Y.C.&H.R.	376	528	0	0	0	0	0	376	32–64	376	0	48	6624	54
N.Y.C.&St.L.	2	..	0	0	0	0	0	2	60	0	0	0	64	0
N.Y., N. H. & H.	330	2103	0	0	0	3	110	113	30	330	0	36–60	7280	270
	..	..	..	..	..	..	..	178	60	..	..	..	..	..
N. P.	667	492	0	0	53	658	64	9	64	0	0	0	28621	0
N. R. of Mex.	36	..	0	0	0	0	0	31	32	9	22	5	592	0
N. & W.	77	382	0	0	0	0	0	13	30	13	30	64	0	1600
O. S. L.	84	198	1	32	6	82	64	1	32	282	82	23	480	0
O. R. & N.	56	55	0	0	7	55	64	1	60	131	0	0	436	0
Pullman	2400	1864	6	0	0	0	0	2400	30	0	0	0	38400	0
P. & L. E.	4	134	1	60	0	0	0	2	30	0	0	0	96	0
	..	..	1	30	..	..	..	..	..	..	..	..	..	..
P. M.	102	259	0	0	0	0	0	102	32	102	0	0	1632	0
St. L. & S. F.	246	388	0	0	0	0	0	236	32	0	76	10	3820	0
St. L. & S. W.	15	208	0	0	0	0	0	4	60	0	11	0	304	0
	..	..	..	..	..	..	..	11	30	..	..	..	..	..
So.	258	..	0	0	0	0	0	228	30	0	0	30	4128	0
S. P.	145	1400	2	0	0	71	64	72	32/64	145	0	0	2200	0
S. P., L. A. & S. L.	76	97	1	0	0	55	64	20	32/60	24	0	0	608	0
T. & N. O.	76	533	0	0	0	0	0	16	64	16	47	0	512	0
T. & B. V.	16	12	0	0	0	0	0	16	32	0	0	0	176	125
U. P.	221	346	0	0	32eng	202	63	13	64	221	0	44	2020	64
	..	..	..	..	..	..	..	6	32	..	..	..	..	..
W. & L. E.	4	72	3	0	0	0	0	1	..	4	0	0	0	0
W. P.	86	..	0	0	0	0	0	85	32	0	0	0	0	0
	..	..	..	..	..	..	..	1	64	87	..	.	1392	..
Penna. East	1092	2310	845	60	0	0	0	19	60	1226	0	489	53632	2550
	..	..	..	..	..	..	..	228	30	..	..	..	..	..
Penna. West	581	717	526	60	0	0	0	5	30	581	0	54	22592	50
	..	..	..	..	..	..	..	50	60	..	..	..	..	..
Vandalia	53	116	42	60	0	0	0	11	60	53	0	0	2128	0
Wabash	..	..	..	..	..	..	..	..	..	..	..	..	..	..
Total, June 30, 1912	13786	29275	1680	..	307	2069	..	7060	..	7847	1330	1849	242932	8817
Total, June 30, 1911	11017	33634	1372	..	192	3185	..	5900	..	..	..	..	202744	..

demand is concentrated upon 15 and 50-watt lamps. Spherical bulbs are used almost exclusively, $3\frac{3}{4}$ in. in diameter for the 50-watt lamp and 2 5/16 in. in diameter for the smaller sizes.

60 The voltage of the lamp is necessarily dependent upon the voltage of the generating system. An investigation of the records of the principal railroads shows the use of the following voltages at the present time: 24, 26, 30, 32, 60, 64 and 110. The bulk of the demand for train lighting lamps is concentrated on two voltage groups, the 30 volt, which includes lamps between 25 and 34, inclusive, and the 60 volt, which includes lamps of from 50 to 65 volts, inclusive.

61 From an engineering standpoint, it is highly desirable that as many of the voltages, wattages and sizes of bulb as is possible be eliminated. From a manufacturing standpoint, it is desirable that some variation be allowed in the voltages, as a large number of high-voltage and low-voltage lamps are accumulated in the manufacture of lamps of a fixed voltage. This brings up the question of the inspection of lamps before purchase.

62 On account of the method of manufacture of train lighting lamps, manufacturers have found it most convenient to inspect the lamps for initial variation by holding the lamps at a fixed candle power, allowing the voltage and current to vary. From the railroad standpoint, this method of inspection seems to be wrong, as the voltage of the circuit is maintained at a very closely fixed value in actual service. If the manufacturers could be persuaded to do so, it would be much more desirable to have lamps inspected by holding them at a predetermined voltage, allowing the variations to take place in the wattage and candle power.

63 According to the present method of inspection, a variation of 5 per cent above and below in voltage and 8 per cent above in watts is allowed. If the demand is concentrated too greatly upon one type of lamp, it will probably be necessary to increase these limits. On the other hand, if the railroads among themselves can standardize their voltages close enough for each voltage class to permit the use of any of the lamps on any railroad, the demand for any one voltage lamp will probably not be great enough to make the increasing of the limits of inspection above mentioned necessary.

64 Until recently little attention has been given to the de-

velopment of reflectors adapted particularly to railroad service. There are several very efficient reflectors now on the market which in some cases have increased the intensity of the light on the reading plane in a passenger car as much as 65 per cent without changing the current consumption or the candle power of the lamps. To secure the best results, attention must be given to the proper location of the filament relative to the reflector.

65 The question of sockets also plays a very important part in car lighting. The socket must be sufficiently rugged in its construction to withstand the jar and vibration encountered in the handling of passenger equipment cars. It must also be so constructed that when the lamp is screwed home there will be sufficient spring in the inner shell and contact to lock the lamp in position, holding it so tight that it will not become unscrewed by the jar and vibration.

STANDARDIZATION

66 The Association of Railway Electrical Engineers has made recommendations as to the standardization of many of the details pertaining to electric car lighting and a number of the railroads of the country have already indicated their intention of following these recommendations by changing their equipment to conform to the recommended practice.

67 The standard voltages recommended by the Association of Railway Electrical Engineers are:

60 volts (nominal) for straight storage and head-end systems

30 volts (nominal) for axle generator systems

68 The lead battery has been fairly well standardized and the construction recommended includes a two-compartment lead-lined tank with rubber jars. The principal variations from the standard in the lead batteries are that some roads are now using lead covers in place of the rubber covers, this change being made to lessen the danger to battery repairmen on account of gas explosions, and the difference in the method of making the connections to the battery posts.

69 The differences in batteries are not of such a character as to preclude the interchange of batteries on different roads.

70 The nickel-iron battery is manufactured only by one company and therefore the battery is more easily obtained to standard construction and dimensions.

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EFFICIENT PRODUCTION OF CYLINDRICAL WORK

By C. H. Norton

ABSTRACT OF PAPER

Early grinding machines were designed for the purpose of producing a higher degree of refinement than was possible with the lathe, both being used as tools of refinement. Later developments show that the highest efficiency is secured in the great majority of cases by using the lathe solely as a roughing tool, transferring the entire responsibility for refinement to the grinding machine.

The process of roughing is not to be considered in the usual sense of turning to a few thousandths of an inch over the finished size, since this involves an element of refinement and loss of time. The workman at the lathe should use the coarsest feeds possible without regard to finish, calipering the work at the beginning of a cut only, where the tool should be fed by hand for a short distance for this purpose. While a piece roughed out in this manner may require a longer time for grinding than if roughed out more carefully with a certain degree of refinement, the combined time for lathe and grinding machine will be less than if part of the refinement was done by the lathe.

Several examples are quoted by the author of work which has been ground in accordance with the principles outlined. It is held that lathe departments and grinding departments should be conducted under one head, instead of separately as is usually the case, to avoid the tendency of the lathe department to consider its work as a finished product, and of the grinding department to insist on careful turning in order to make a better showing of time required for grinding.

EFFICIENT PRODUCTION OF CYLINDRICAL WORK

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Non-Member

Turning is one of the oldest of the mechanic arts, and the lathe is one of the oldest metal working machines; but the lathe has been a very inefficient tool until the present time. We now have lathes so designed that they can be made efficient. I believe, however, that we are not as a rule using the modern high power lathes as we should, and are losing what they were designed to save, viz., *time;* that while we use them as roughing machines we also use them as refining machines.

2 Most cylindrical work must finally have size within small limits and be truly cylindrical within still smaller limits, which led mechanics in the past to study the art of refined turning. When this art had reached a high degree of refinement, men discovered that it did not satisfy them and cylindrical grinding was introduced to give more refinement than could be obtained with the lathe. The natural conclusion at that time was that the grinding machine should take up the work of refinement after the limit of refinement had been reached with the lathe and the early grinding machines were, therefore, designed with this thought in view.

3 Later, however, it was shown that ground cylindrical work could be produced at a cost no greater than the turning alone had cost, simply by using the lathe to remove a large proportion of the metal necessary to be removed and not at all as a means of refinement; transferring the entire responsibility for refinement to the grinding machine.

4 While some understand this and are producing cylindrical work efficiently, I believe the great majority are still under the

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spell of tradition, and are turning what they call "good work:" work that shows care and skill on the part of the operator and accuracy in the lathe. Such care and skill with turned work is even more senseless today than the old time practice of shaping acorns on all screw heads and putting cast iron eagles on the tops of machine tools, because in case of the former the grinding machine removes all trace of the lathe's accuracy and the operator's skill in a moment of time, while the latter remained for years to gratify the taste of those who had an eye for such artistic embellishment.

5 The developments of the last few years have brought about the facts that, except in rare cases, efficiency in the production of cylindrical work means, (*a*) the use of the lathe as a roughing tool only, and (*b*) the modern grinding machine for refinement.

6 Most lathe operators and most foremen seem to understand roughing to be simply the turning of work a few thousandths over the finish size, when in reality they are not roughing at all, but simply turning to a certain degree of refinement with the notion that by so doing they are (*a*) doing a creditable job, and (*b*) that they are helping the grinding machine. It is difficult to secure roughed work from lathe operators, i.e., work that is not in any sense refined.

7 Tradition impels nearly all to consume much more time than would be necessary if the work were merely roughed, roughing being confused with a certain degree of refinement. Roughing is not refinement of any degree, and refinement does not necessarily mean perfection. There are many degrees of refinement, but never degrees of accuracy or perfection. Accuracy and perfection are final and fixed, they have no limits.

8 Those who would produce cylindrical work efficiently must recognize the fact that different cases require different degrees of refinement, and whether the work requires a very low degree of refinement or a degree closely approaching accuracy, the lathe work, to be efficient, should be the same, viz. as *rough* and as *cheap* as it is possible to make it.

9 The lathe today should be a roughing tool only, and the grinding machine a refining tool only, not a perfecting tool. Some grinding machines may produce greater refinement than others, but none can produce literally exact and perfect cylinders. While it is true many mechanics look upon a high degree of

refinement as accuracy and speak of it as such, it is easy to prove that it is not accuracy, but simply a high degree of refinement. When all recognize these facts there will be greater efficiency in the production of cylindrical work.

10 Because these facts about roughing and refining are not well understood and men do not combine the lathe and grinding machine in a well defined and efficient manner, the machine industry and railroads are losing large sums each year and I will try to show where some of the loss occurs.

11 We are losing because we turn closer than *about* 1/32 in. above the finish size. I use the word *about* because we lose by turning carefully. To require lathe work turned to thousandths today is like burning money for the fun of it. We are losing by allowing workmen to use any but the coarsest feeds possible in each case, even to the use of the screw cutting gears to obtain them.

12 Feeds as coarse as four to the inch should be used in some cases and six, eight and ten should be quite common. When work is too frail to allow such feeds with one cut, then another, or even three cuts should be made at faster cutting speed; for, while it is true that very coarse and imperfect turning will require longer to grind than close, careful turning, the combination of coarse, cheap turning with grinding will produce cylindrical work more efficiently. The modern grinding machine will remove metal most efficiently when that metal is in the form of coarse screw threads.

13 High ridges do not increase the cost of production. We lose money when workmen are allowed to caliper cylindrical work on the tops of the ridges, as nearly all do. This accounts for the misunderstanding about coarse turning and the allowance for grinding. Calipering on the tops of the ridges makes it necessary to turn with relatively fine feed to enable clean grinding. To avoid this some workmen use a broad nose tool with the coarse feed which necessitates slower cutting speed because more metal is removed than with a pointed or grooved cutting tool. Ridges left by a more pointed tool do not increase the cost of producing that work, but decrease it.

14 It is so simple to feed by hand a slight distance at the end of the work, where the calipering can be done correctly, however coarse may be the power feed later, and however high

the ridges. Money is lost by allowing workmen to measure roughed work elsewhere than at the end; hours are wasted by operators calipering at several points and then because of spring or tool wear, resetting and turning that portion again to secure straight work. They forget that the modern grinding machine was designed to do all of the straightening and all of the sizing, regardless of what the errors of roughing may be. They forget, or have never known, that the art of producing cylindrical work has progressed beyond that stage where it was necessary for the lathe to do its best before the grinding machine assumed the work of refinement.

15 The modern grinding machine is literally a correcting machine. The roughing can literally be rough, and roughness does not mean an approximate refinement. It is easy to figure the saving in time of turning a piece of work 6 in. in diameter by 6 ft. long when the last cut before grinding is made with a feed of six per inch, instead of 32 per inch which is very common. When it is considered that the grinding machine will grind this six per inch work to the finish size complete in less time than is required to file 32 per inch work, we see the great possibilities for saving cost with the lathe, if combined with a modern grinding machine.

16 In some cases money is lost by roughing at all in the lathe. Please note that I say *in some cases*. Efficient production of cylindrical work is usually accomplished by roughing first in the lathe; but there are cases when the difference in favor of grinding without first roughing in the lathe is very great.

17 The most important study for those who would secure efficient production of cylindrical work is the roughing of that work. It is here that the greatest possibilities for saving lie. If all would make a careful study of the roughing preparatory to grinding instead of trying to improve upon the methods of grinding advocated by grinding machine makers, there would result a more efficient production of cylindrical work and a clearer view of the real reasons why the grinding machine maker advocates certain methods, and why the modern heavy grinding machine was introduced.

18 At a works where large numbers of cylindrical pieces are manufactured $1\frac{1}{8}$ in. in diameter and about 10 to 11 in. long, 5 minutes was required to turn each, removing about 1/16 in. from

the diameter using a high-speed steel tool and revolving as rapidly as the tool would stand. The plan was to turn close to the finish diameter and care was taken to secure straight, smooth and round work to save time when grinding, because they considered the grinding expensive, and that this expense must of course be added to the cost of turning. The grinding in this case was 1 minute. The total time for producing these pieces was 6 minutes.

19 A change was made in the shape of the tool point and the size limit for turning increased. The traverse feed was increased, the cutting speed remained unchanged and the work was turned in 1 minute each, while the grinding time was doubled, viz. 2 minutes; the saving was 3 minutes, or one-half of the original time.

20 Some idea of what is lost by allowing lathe work to be turned straight, smooth and close to size before grinding may be obtained from the following illustration of actual work on some forgings about 4 ft. 6 in. long, turned portion to finish 2 in. plus or minus 0.0005 in. in diameter, and 3 ft. long. The lathe operator turned these in the same manner that the majority of operators are now turning such work, viz. as such work was turned before grinding machines were introduced, except that 0.010 in. was left on the diameter for grinding. The turning required two cuts, as there was more than 1/8 in. to remove and the work was somewhat frail as well as irregular. The first cut was considered by the workman a very coarse heavy cut; the second was like the last cut in the majority of shops, viz. it gave what the lathe men and most foremen call "a good job." When ready for grinding the time consumed by turning was a total of 25½ minutes. This appears to be too slow, but the forging was too slim to allow little, if any more, with one rough cut, and the lathe was an old-time weak power frail machine. The grinding time on this was 3 minutes, making a total of 28½ minutes.

21 When tradition was ignored and the work was made ready for grinding in the proper way, the following saving was accomplished: Two roughing cuts were taken, because (*a*) the lathe was not powerful, and (*b*) the work was frail. The feed was ten per inch, cutting deep grooves; the time complete ready for grinding was 9 minutes. The work was not straight, and was what lathe men would style "a very poor job." The grind-

ing time to the exact limits was 9 minutes, or three times longer than with the more careful turning; a total of 18 minutes, a saving of $10\frac{1}{2}$ minutes on each forging.

22 An attempt to rough this work with one cut instead of two resulted in a loss of 2 minutes. A modern lathe having more power with the direct belt would have carried the same feed and depth of cut at higher cutting speed with a still further reduction of time. This shows what may be accomplished with weak low power lathes.

23 To install high power lathes and continue to turn just as smoothly and accurately as of old, simply allowing a few thousandths for grinding, would seem the height of wastefulness. Whether high power lathes or old low power lathes are used mechanical industries are losing an enormous sum each year because of the lack of understanding of the real problem affecting the efficient production of cylindrical work, which is the proper combination in each individual case of rough turning and grinding.

24 Some idea of what we are losing by not wearing our "thinking caps," is shown by the following: Some plain shafts, 1 5/16 in. in diameter, 62 in. long were to be made to ordinary limits of size and finish. Tests were made to determine whether it was better first to turn these or to grind direct from the rough bars. The turning required 6 minutes ready for finish grinding, while to grind off the same amount as was turned off required 9 minutes. A number of mechanics pronounced it cheaper to turn the shafts first. "Of course." But in this case after turning in 6 minutes, each shaft required straightening before grinding, which consumed 10 minutes, totaling 16 minutes. When the roughing was done by grinding instead of turning no straightening was necessary, and the roughing cut was removed in 9 minutes.

25 In this case, therefore, the turning was actually lost. There may be similar cases where the shafts will require some little straightening even if roughed by grinding, so that each case should be investigated instead of making a rule that all must or must not be turned first. Usually, however, no straightening is necessary when the roughing is done by grinding.

26 Money is being lost because the majority of designers have not kept pace with the development along these lines. A

well-known runabout motor car has a crankshaft, which if made about 1 in. longer over-all could be produced for 50 cents less on each crank. When it is considered that about 30,000 of these cranks have been made, representing a possible saving of $15,000, we see the importance of considering the combination of lathe and grinding work.

27 Fig. 1 shows a design of crankshaft that is expensive to machine, since it requires carefully finished lathe work. It will be seen that this shaft has a flange that must be finished complete with the lathe. Aside from the flange also the design necessitates lathe finishing of the majority of the work. Grinding is a small part of the whole.

28 Fig. 2 shows a design that requires no lathe work whatever, there being clearance for the sides of the grinding wheel at all points. The illustration shows the six operations necessary to complete the shaft after cutting off the ends and centering.

29 These shafts are finished to the usual limits directly from the drop forging by grinding. The net labor cost for cutting off ends, centering, and machining complete with one keyway and the thread ready for the automobile, is 55 cents. The cost for grinding wheels is $2\frac{1}{2}$ cents each; cost for diamonds to true and shape the wheel, is 2 cents each.

30 Another feature affecting the cost of production is the matter of ordinary shoulders. This is overlooked by many designers and practically all draftsmen. It is easy to draw two sharp lines that cross each other and form a sharp corner and the fact that it costs less to provide room for the shoulder fillet on the other member than to make a sharp corner in the cylindrical member is lost sight of. This no doubt is the result of the experience of the past when there were no powerful grinding machines to locate and form such shoulders cheaper than the lathe. Shoulders to be made with the grinding wheel require a slight fillet.

31 When numbers of duplicate pieces are to be produced and a small fillet is allowed, not only can the cylindrical portion be roughed with the lathe and not at all refined, but the shoulders can also be roughed, because the grinding machine, when required, is provided with a locating bar. By the use of this bar, the exact location of all shoulders can be secured by grinding and without measuring. At the same time the cylindrical

portion can be ground with the same grinding wheel. The roughing in the lathe can be done with the same tool that roughs the cylindrical portion because the lathe in this case is not required to give to the shoulder any particular shape. When it is found best to *locate* and *finish* shoulders with the lathe because

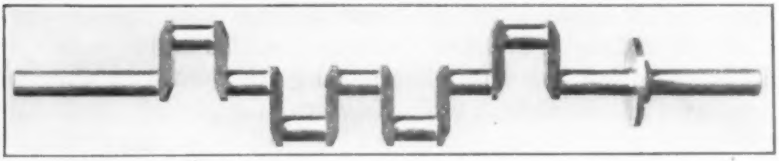
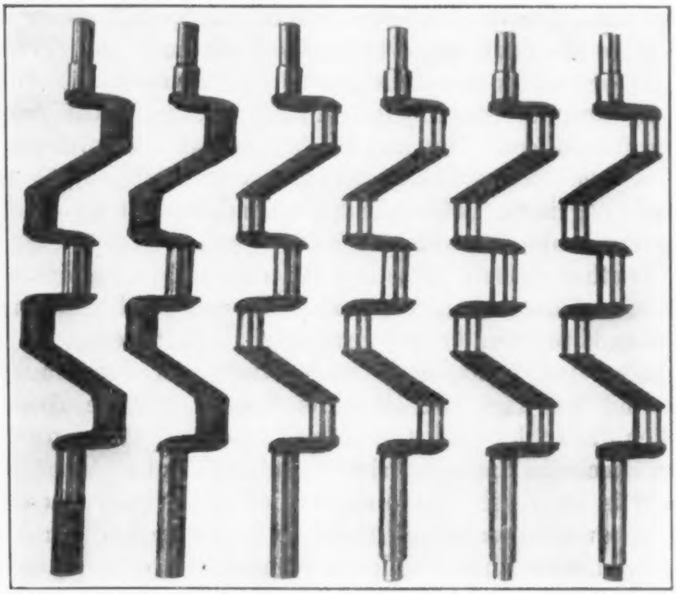


Fig. 1 Example of Expensive Design



1 2 3 4 5 6

Fig. 2 Examples of Inexpensive Design

there is no locating bar for the grinding machine, it should not be necessary to neck in at the shoulders for grinding. This is an old and out-of-date notion, for, with a shoulder cut sharp by the lathe tool, the grinding does not materially change the corner.

32 A more efficient method when no locating bar is at hand

is to rough the work in the lathe, leaving the shoulders about 1/32 in. long and whatever angle the roughing turning tool may form. Next grind the cylindrical portion and at the same time cut out the angle at the shoulder with the grinding wheel. The

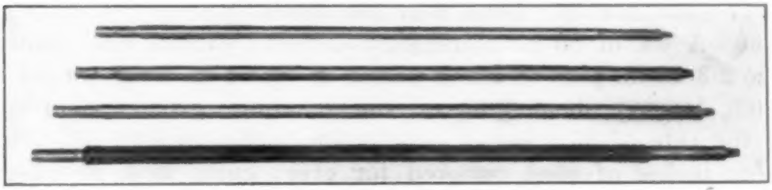


Fig. 3 Work too Slim to Turn before Grinding

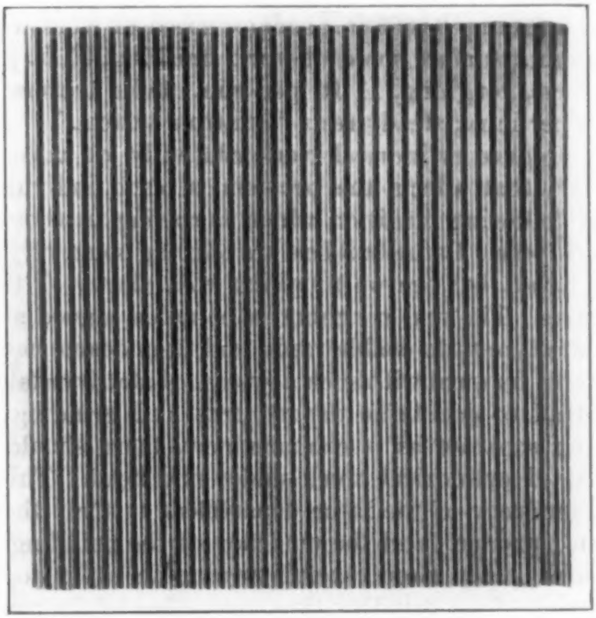


Fig. 4 Work Ground without Turning

work should then go to a lathe where a more skillful operator can locate the shoulders exactly; if a sharp corner is really necessary it can then be made.

33 Fig. 3 shows work so slim that turning of the main portion is out of the question if it is to be produced efficiently. The

main or center portion should be ground direct without turning.

34 Fig. 4 shows 37 slim shafts ground direct from the black stock. To turn them first would be a waste of time. They were 30 in. long, 11/16 in. in diameter in the rough, ground to 0.625 in. plus or minus 0.0005 in.; handled twice and ground complete in 17 minutes each. These were tool steels.

35 A lot of 60 countershafts, 35 point carbon steel, finish size 2 3/16 in. plus or minus 0.0005 in. by 66 in. long, turned 8 pitch, leaving about 1/32 in. for grinding; ground complete at the rate of 35 minutes each; cubic inches ground off, 278; cubic inches of steel removed for every cubic inch of wheel wear, 20.7; total cost of wheel for 60 shafts, 38 cents; total cost of power, 35 cents; wheel used, 24 combination grade L alundum; radial depth of cut, when roughing, 0.001 in.; radial depth of cut, when finishing, 0.0005 in.; surface speed of work, roughing, 42 ft. per min.; surface speed of work, finishing, 35 ft. per min.; table traverse, roughing, 11 ft. per min.; table traverse, finishing, 10 ft. per min.; steady-rests "multiple system."

36 To produce cylindrical work efficiently we must make a study of all that affects the problem in each individual case instead of following tradition which is costing us thousands of dollars each year. Lathe builders are offering some very efficient roughing lathes, but are we roughing with them? I think not in most cases. The real roughing lathe needs a good system of multiple steadyrests to enable rapid and deep coarse cuts. Will someone come forward with such a system of steadyrests?

37 Instead of our lathe departments and grinding departments being separate as is common now, there should be one department of cylindrical work under one head. This would check the tendency of the lathe department to treat their work as a finished product, and the tendency of the grinding department to insist upon close careful turning in order to make a better showing of the grinding time, regardless of the real cost of production.

38 There is much yet to be learned about the preparation for grinding of cylindrical work; no two pieces of work require the same lathe treatment. There is a great need for thinking men as foremen and operators. Such men can effect great savings by working out in each case the best combination of turning and grinding of cylindrical work.

THE STRENGTH OF GEAR TEETH

By Guido H. Marx

ABSTRACT

The experiments recorded in this paper were made with a view to securing data upon the allowable unit fiber stress for cast iron gear teeth when in operation. The gears tested were all of 10 diametral pitch, $14\frac{1}{2}$-deg. involute, $1\frac{1}{16}$-in. face. Comparison is made of these results and of those given by the use of the Lewis formula, together with a discussion of such discrepancies as were found to exist. Factors such as are of action and width of face were found to exert an influence in ways not usually taken account of. Practical suggestions are given upon the proportions of gears based upon the observed manner of breaking of the various gears tested.

THE STRENGTH OF GEAR TEETH

BY GUIDO H. MARX, STANFORD UNIVERSITY, CAL.

Member of the Society

The teeth of gear wheels when transmitting power are individually subjected to an action akin to that applied to a beam fixed at one end, with a load somewhere between the fixed and the free ends. All standard formulae or diagrams for the proportioning of such teeth therefore involve a factor representing the allowable unit fiber stress in a cantilever beam subjected to a bending moment.

2 The experiments described in this paper were undertaken with the primary purpose of throwing some light, if possible, upon the question of this allowable unit fiber stress for cast-iron gear teeth under operating conditions, since definite data upon this point have been lacking, particularly with reference to the effect of pitch line velocity. To this vital lack, attention has been called repeatedly by writers upon the subject of gearing.

3 Thus, in Wilfred Lewis's well-known Investigation of the Strength of Gear Teeth[1] occurs the following statement bearing upon this matter:

> What fiber stress is allowable under different circumstances and conditions cannot be definitely settled at present, nor is it probable that any conclusion will be acceptable to engineers unless based upon carefully made experiments. In the article referred to[2] certain factors are given as applicable to certain speeds and, in the absence of any later or better light upon the subject, Table II has been constructed to embody in convenient form the values recommended.

[1] Proceedings of the Engineers' Club, Philadelphia, 1893, vol. 10; also American Machinist, May 4, 1893.

[2] Power Transmitting Mechanism: On the Strength of the Teeth of Wheels, J. H. Cooper, Journal of the Franklin Institute, 1879.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 29 West 39th Street, New York. All papers are subject to revision.

TABLE II SAFE WORKING STRESS S FOR DIFFERENT SPEEDS

Speed of Teeth, Ft. per Min.	100 or less	200	300	600	900	1200	1800	2400
Cast Iron	8000	6000	4800	4000	3000	2400	2000	1700
Steel	20000	15000	12000	10000	7500	6000	5000	4300

It cannot be denied that slow speeds admit of higher working stresses than high speeds, but it may be questioned whether teeth running at 100 feet a minute are twice as strong as at 600 feet a minute, or four times as strong as the same teeth at 1800 feet a minute. For teeth which are perfectly formed and spaced, it is difficult to see how there can be a greater difference in strength than the well-known difference occasioned by a live load or a dead load, or two to one in extreme cases. But, for teeth as they actually exist, a greater difference than two to one may easily be imagined from the noise sometimes produced in running, and it should be said, that this table is submitted for criticism rather than for general adoption.

4 It is very evident that Mr. Lewis only offered the values of his Table II tentatively. A careful examination of Mr. Cooper's paper fails to disclose any table from which Table II is immediately transferred, but on page 15 of that paper will be found a series of factors, credited to E. R. Walker, New Castle-under-Lyme, 1868, varying with the rim-speed of wheels, by which the "breaking load of tooth" is to be divided. These factors are given as follows:

$m =$ 3 for very slow speed without shock.
$=$ 4 when rim of wheel runs 3 ft. per sec.
$=$ 5 when rim of wheel runs 5 ft. per sec.
$=$ 6 when rim of wheel runs 10 ft. per sec.
$=$ 8 when rim of wheel runs 15 ft. per sec.
$=$ 10 when rim of wheel runs 20 ft. per sec.
$=$ 12 when rim of wheel runs 30 ft. per sec.
$=$ 14 when rim of wheel runs 40 ft. per sec.

It will be seen that these values correspond to Mr. Lewis's table if 24,000 and 60,000 lb. per sq. in. are used for the ultimate fiber stress in flexure of cast iron and steel respectively. This is an apparent oversight on Mr. Lewis's part. While 24,000 lb. per sq. in. is a legitimate assumption for the unit strength of cast iron in tension, a value of at least 36,000 may be taken ordinarily for the modulus of rupture for transverse stress or ultimate unit stress in outer fiber, due to bending. That Mr. Cooper meant this modulus of rupture by "breaking stress" is obvious from pages 8 to 11 of his paper. He also quotes J.

Christie as using a modulus of rupture of 36,000, with a factor of safety of never less than 4, and then under the most favorable conditions, as well-fitted gears, rigidly supported, running at moderate speed, and stress evenly distributed. For strains suddenly applied the factor of safety should be 6, and when accompanied by severe shocks and sudden reversions of strains it should be 8. With these assumptions, for cast-iron teeth, Mr. Christie would allow for safe working stresses from $\frac{36000}{4} =$ 9000 lb. per sq. in., down to $\frac{36000}{8} = 4500$ lb. per sq. in., as contrasted to a range in Table II of from 8000 lb. down to 1700 lb.

5 In view of the fact that Mr. Christie gives several examples from practice to sustain his position, and of Mr. Lewis's own doubt as expressed in the foregoing quotation from his paper, of so wide a range being justified as that called for by his Table II, and of the further fact that this table was based upon a modulus of rupture much too low, it seems strange that these tentative values have been so generally accepted; and the need of adequate experimental data becomes apparent.

6 Mr. Cooper on page 9 of his paper says:

> It is certainly necessary to know the strain that breaks a piece of projecting cast-iron of given size and shape, when the stress is laid on quietly, as well as when it is driven on with considerable velocity.
>
> It is also important to know the fractional part of the rupturing weight or stress which may be repeatedly laid on with perfect safety, to insure the continuance of adhesion under the usual conditions of working. Upon these essential features of each case, the criteria, almost every engineer's guide book is silent.

7 A copy of Brown & Sharpe's Treatise on Gearing which the writer has had since 1892 gives the only data he recalls having seen of cut gears tested to rupture under running conditions. On page 118 is the following:

> We give a few examples of average breaking strain of our Combination Gears, as determined by dynamometer pressure taken at the pitch line.

Diametral Pitch	Face	No. of Teeth	R. P. M.	Pressure at Pitch Line
10	1 1/16	110	27	1060
8	1 1/4	72	40	1460
6	1 9/16	72	27	2220
5	1 7/8	90	18	2470

If we take a safe pressure at $^1/_3$ of the foregoing breaking strain we shall have

for 10 Pitch $353^1/_3$ lb. at the pitch line
8 Pitch $486^2/_3$ lb. at the pitch line
6 Pitch 740 lb. at the pitch line
5 Pitch $823^1/_3$ lb. at the pitch line

8 In order to add to the available data concerning the strength of modern, cut, cast-iron gear teeth under operating conditions, the writer had the apparatus shown in Fig. 1, constructed. It consists of a base-plate carrying three adjustable bearings. Only one of these is clearly seen in the immediate foreground of the illustration, but at its right can be seen a corner of a second, while the third is hidden by the brake wheel. The action of the apparatus, reduced to the utmost simplicity of construction, can be seen by reference to Fig. 2.

9 The motor, a 25-h.p. 220-volt, direct-current machine, is connected by a Morse chain to shaft No. 1. Shafts No. 1 and No. 2 are connected by a pair of cut-steel change gears of 8 diametral pitch which can be varied to give a wide range of velocity ratios. Shafts No. 2 and No. 3 are connected by the cast-iron, cut gears to be tested. On the outer end of shaft No. 3 there is a flanged, water-cooled brake wheel carrying a prony brake. The arm of the brake rests, by means of a steel knife edge and plate, on a platform scale. Each bearing is provided with a sight feed lubricator. The measurement of the efficiency of transmission not being the object of the experiment, the friction of the bearings is neglected. The only effect of this is to make the computed breaking load of the teeth a very little less than its real value in each case. By means of the slotted baseplate and the tongues on the bottom of the bearings, the latter can be slid into place, the gears accurately meshed without binding or backlash, and then securely held by means of two square-head bolts to each bearing. The Morse chain is lubricated with

a graphite and grease mixture, and the gears, both steel and cast-iron, with ordinary thick grease lubricant which was also freely used on the brake. Before starting each run a wooden guard was placed over the gears to be tested. That this was not a useless precaution was indicated by the fragmentary condi-

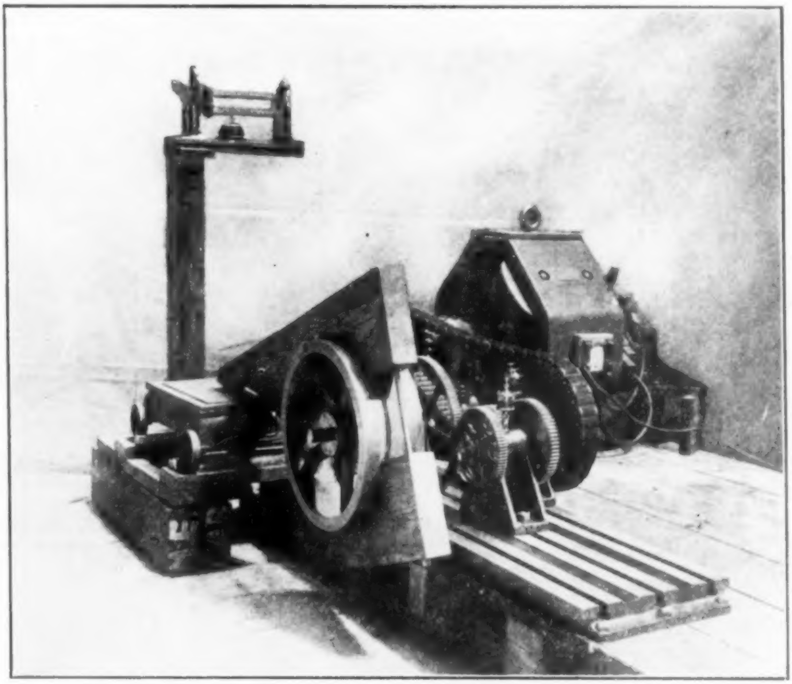


FIG. 1 GENERAL VIEW OF TESTING APPARATUS

tion of the gears, particularly those having arms, at the conclusion of many of the runs.

10 The tests were made in the mechanical laboratory of the Leland Stanford Junior University. For each run the motor was started with zero load applied by the prony brake. The scale weights were then set at the lowest load and the brake tightened until the scale beam floated. Simultaneously the rate of rotation was observed with a tachometer which was calibrated upon completion of the tests and the necessary slight corrections made in the computed results. Calibration of the scales showed them to be entirely correct throughout the range used.

Increments of load on the scale began at 5 lb. each while the load was low, and were diminished to two lb. and one lb. as the probable breaking load was approached. The unexpectedly high breaking strength shown by the teeth[1], particularly at the higher velocities, made it impossible to break the gears at pitch speeds much exceeding 500 ft. per min. with this apparatus.

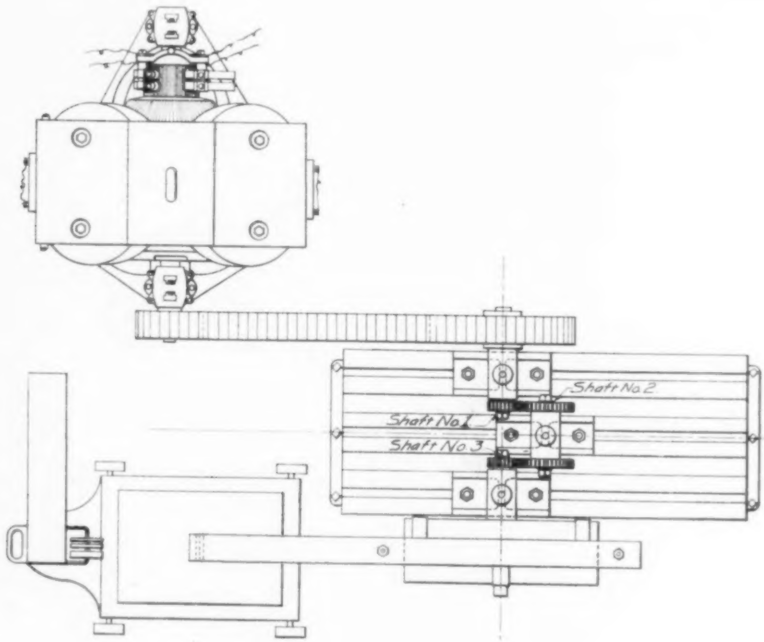


Fig. 2 Plan of Apparatus

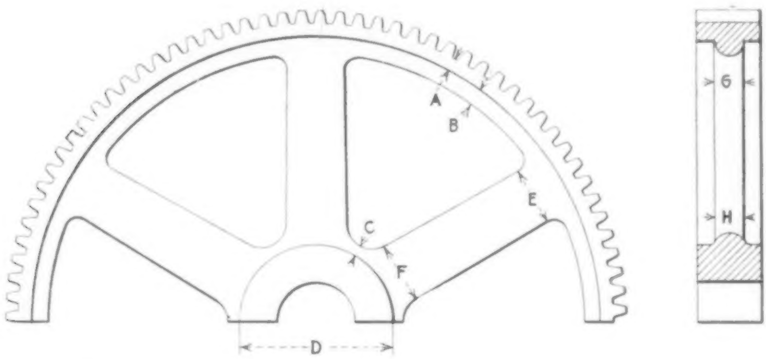
At the higher speeds the teeth were capable, without rupture, of transmitting all of the power the motor could develop.

11 The gears tested were all 10 diametral pitch, cast-iron, 14½-deg. involute, purchased of the Brown & Sharpe Mfg. Co. without intimation of the special purpose for which they were intended, and were of the ordinary stock proportions. The width of face in all cases was 1 1/16 in. The 20- and 30-tooth gears were solid, the 40-tooth webbed, and the others each had six arms. The twenties and thirties had a bore of 1 1/16 in.

[1] Preliminary calculations had been based upon the Brown & Sharpe figures quoted in Par. 6.

and a ¼ in. keyway. The rest had a bore of 1 5/16 in. and a 5/16 in. keyway. The general proportions of the arm gears are shown in Fig. 3.

12 As shown by the logs, the main tests were conducted in two series; the first being made in 1911 with single observations under each set of conditions; and the second, made in 1912, being planned to cover the ground more completely, with the further intention of making three runs in each case under identi-



	A	B	C	D	E	F	G	H
60 T	1/4	1/4	1/4	2 7/16	7/8	13/16	3/8	3/8
80 T	9/32	1/4	1/4	2 7/16	7/8	15/16	3/8	3/8
100 T	5/16	3/16	3/16	2 1/2	7/8	1	7/16	15/32
150 T	3/8	9/32	5/16	2 1/2	31/32	1 3/16	1/2	17/32

Note—All dimensions are in inches. Bore 1 5/16 in. Face 1 1/16 in. in all cases

FIG. 3 DIMENSIONS OF ARMS OF GEARS TESTED

cally the same conditions in order to eliminate errors of observation and of variable material.

13 Fig. 4 shows graphically the results of all the foregoing tests involving 20-tooth pinions. Abscissae are pitch-line speeds in feet per minute, and ordinates are forces at the pitch point equivalent to the breaking loads in pounds. This equivalent breaking load is, of course, equal to the net brake load at rupture by brake arm divided by the pitch line radius of gear on brake shaft. Curve A is drawn representing the results of all tests with 20- and 30-tooth gears in mesh. As there were several observations taken for each pitch speed, the numerical average

TABLE 1 TESTS OF 10 DIAMETRAL PITCH, CAST IRON, 14½ DEG. INVOLUTE GEARS
LOG OF FIRST SERIES OF RUNS

Test No.	Date, 1911	Gears Tested 10 Pitch, C. I.		Sprocket r.p.m. Observed Tachometer	Brake r.p.m. Observed Tachometer	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Intermediate Shaft		
01	June 5	20	150	...	260	120	20	110	Neither gear broken. Fuse blew and motor stopped. Intermediate shaft heated and seized; driven out, draw-filed; bearing scraped. The 5/16 in. machinery steel key of 120 T–8 pitch, steel change gear sheared off. Shaft replaced by new one in subsequent tests. Both 5/16-in. keys replaced by 1/2 in. This was trial run to test out apparatus.
02	June 9	20	100	...	375	120	20	125	Neither gear broken. Limit of motor power. Fuse blew.
03	June 9	20	100	355	...	20	120	240	Broke 3 teeth at root in 20–T pinion. Broke 100–T gear in 4 pieces. No teeth broken out of large gear, but rim broken.
04	June 9	20	30	353	...	20	120	71	Broke as load was being increased from 70 to 72 lb. Broke 20–T pinion in 3 pieces, each running to the bore; no teeth broken at root. Apparently gave way at key-way. Broke 1 tooth at root in 30–T gear.
05	June 10	20	30	316	...	70	70	50	Broke 13 teeth at root in 20–T pinion. Also broke at key-way and into 3 pieces. Broke 8 teeth out of 30–T gear. Battered and loosened 1/4-in. key on intermediate shaft. Both 1/4-in keys replaced by hardened tool steel, same size. Probable that break occurred by "canting" of loosened key.
06	June 10	20	30	...	360	70	40	60	Broke 20–T pinion in 4 pieces, one at key-way. One tooth cracked at root. No break in 30–T gear. Broke as load was beginning to be raised from 60 lb. to next scale setting, 65 lb.

TABLE 1—Continued

07	June 10	20	30	...	480	70	30	55	The 30–T gear was the same one used in test No. 06. Broke 20–T pinion in 4 pieces, one being a single tooth with wedge running to key-way. Broke 7 teeth off at root. Also broke 7 teeth off at root in the 30–T gear.
08	June 12	20	30	...	218	70	70	60	This test was made as a repetition of No. 05, as that test seemed unreliable because it pointed to splitting by loose key. Broke 20-T pinion in 2 pieces; also broke out 10 teeth at root and cracked 2. Broke 3 teeth out of 30–T gear.
09	June 12	30	40	...	236	70	70	79	Broke 18 teeth of 30–T gear at root and crushed one from end. Broke 14 teeth at root of 40–T gear. Heard crack at 78 lb. load and stopped run. No teeth found broken. Started again and broke between 78 and 80 lb. load.
010	June 12	40	60	...	206	70	70	148	Broke 9 teeth out of 40–T gear and 9 teeth out of 60–T gear.
011	June 12	40	40	...	230	70	100	104	17 teeth stripped and 7 partially broken on 40–T driver. 10 teeth stripped and 2 partially broken on 40–T driven.
012	June 12	40	60	335	...	70	100	148	9 teeth stripped and 1 broken on 40–T gear. 6 teeth stripped out of 60–T gear.
013	June 12	40	80	326	...	70	100	222	8 teeth stripped out of 40–T gear. Note depth of break at bottom. Broke rim of 80–T gear in 4 pieces.
014	June 12	60	80	328	...	70	100	223	Broke between 222 and 224 lb. load. 21 teeth stripped out of 60–T gear. 80–T gear broken in 6 pieces and 8 teeth stripped.
015	June 13	80	100	315	...	40	100	280	No teeth broken out of either gear. 100–T gear broke at 3 places in rim, also broke out hub and arms at key-way. 5/16-in. key loose in key-way. It looks as if the yielding of the key-way in shaft allowed the key to cant slightly and split the hub. (See also tests Nos. 05 and 016). Key replaced with hardened tool-steel key. Strength of teeth not shown in this test.
016	June 13	30	150	328	...	40	100	380	30–T gear had 7 teeth stripped. 150–T gear very obviously wedged apart from hub outward by the key which was found canted so that one corner of the bottom was 5/64 in. from the bottom of key-way. (See tests Nos. 05 and 015). Large gear cracked at hub and rim, not broken apart.

TABLE 2 TESTS OF 10 DIAMETRAL PITCH, CAST IRON, 14½ DEG. INVOLUTE GEARS
COMPUTED RESULTS OF FIRST SERIES OF RUNS

SPEED CORRECTIONS MADE FOR ERROR OF TACHOMETER; BRAKE ARM = 32 IN.; ZERO BRAKE LOAD = 3½ LB., BALANCED ON UPPER SCALE-BEAM

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min.	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 p. Steel Driver	Intermediate 8 p. Steel Driven	Intermediate 10 p. C. I. Driver	Brake 10 p. C. I. Driven						
01	120	20	20	150	266.0	3.927	110	1044.6	469	Gears not broken. Motor stalled. Run does not count.
02	120	20	20	100	384.0	2.618	125	1005.3	800	Gears not broken. Motor stalled. Run does not count. Demonstrates that teeth are stronger than this.
03	20	120	20	100	12.13	2.618	240	31.76	1536	Reliable run.
04	20	120	20	30	40.22	0.7854	71	31.61	1515	Reliable run.
05	70	70	20	30	215.3	0.7854	50	169.23	1067	Indicates that gear broke as suspected (see Log, Table 1) by canting of key. Repeated in test No. 08
06	70	40	20	30	369.0	0.7854	60	290.03	1280	Reliable run.
07	70	30	20	30	493.0	0.7854	55	387.5	1173	Reliable run
08	70	70	20	30	223.0	0.7854	60	175.38	1280	Reliable run takes place of test No. 05, of which it was a duplicate but with new key
09	70	70	30	40	241.0	1.047	79	252.33	1264	Reliable run
010	70	70	40	60	210.5	1.571	148	330.7	1579	Reliable run
011	70	100	40	40	235.0	1.047	104	246.05	1664	Reliable run
012	70	100	40	60	160.7	1.571	148	252.46	1579	Reliable run
013	70	100	40	80	116.9	2.094	222	244.79	1776	Reliable run
014	70	100	60	80	176.4	2.094	223	369.38	1784	Reliable run
015	40	100	80	100	103.0	2.618	280	269.65	1792	No teeth broken. Shows these teeth are stronger than even this
016	40	100	30	150	26.9	3.927	380	106.64	1621	Indicates further that gear broke (see Log, Table 1) by canting of key

TABLE 3 TESTS OF 10 DIAMETRAL PITCH, CAST IRON, 14½ DEG. INVOLUTE GEARS

LOG OF SECOND SERIES OF RUNS

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tacho-meter	Brake r.p.m. Observed Tacho-meter	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Inter-mediate Shaft		
1	May 4	20	30	420	...	20	120	68	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
2	May 4	20	30	422	...	20	120	78	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
3	May 4	20	30	422	...	20	120	76	20-T pinion broke at key-way; 3 pieces, and 1 tooth broken out at root. Broke 1 tooth out of 30-T gear
4	May 4	20	30	410	...	70	70	62	20-T pinion broke at key-way; 3 pieces, and 14 teeth broken out at root. Broke 5 teeth out of 30-T gear
5	May 4	20	30	414	...	70	70	58	20-T pinion broke at key-way; 5 pieces, and 4 teeth broken out at root. Broke 5 teeth out of 30-T gear
6	May 4	20	30	414	...	70	70	60	20-T pinion broke at key-way; 3 pieces, and no teeth broken out at root. 30-T gear not broken at all
7	May 9	20	30	450	...	100	40	52	Neither gear broke. Fuse blew. Motor stalled. Made new electrical connections at higher voltage
8	May 11	20	30	...	520	100	40	52	Same gears as test 7. 20-T pinion broke at key-way; 5 pieces, and 1 tooth broken out at root. 30-T not broken
9	May 11	20	30	...	540	100	40	58	20-T pinion broke at key-way; 5 pieces, and 1 tooth broken out at root. 30-T gear not broken
10	May 11	20	30	...	550	100	40	58	20-T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30-T gear not broken
11	May 11	20	30	...	660	120	40	54	20-T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30-T gear not broken
12	May 11	20	30	...	660	120	40	60	20-T pinion broke at key-way; 4 pieces. No teeth broken out of either gear. Note strength of teeth indicated

TABLE 3—Continued

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tacho-meter	Brake r.p.m. Observed Tacho-meter	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Inter-mediate Shaft		
13	May 11	20	30	...	660	120	40	52	20–T pinion broke at key-way; 4 pieces, and 3 teeth broken out at root. 30–T gear not broken
14	May 11	20	30	680	...	120	30	48	Did not break gears. Too fast for brake. Abandoned attempt to make runs at this speed
15	May 11	20	30	...	400	70	40	52	Same gears as test 14. Broke before the scales quite balanced at 52 lb. setting. 20–T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30–T gear not broken
16	May 11	20	30	...	396	70	40	54	Same gears as test 14. Broke before the scales quite balanced at 52 lb. setting. 20–T pinion broke at key-way; 4 pieces, and 1 tooth broken out at root. 30–T gear not broken
17	May 11	20	30	...	400	70	40	56	Same gears as test 14. Broke before the scales quite balanced at 52 lb. setting. 20–T pinion broke at key-way; 2 pieces, and 15 teeth broken out at root. 30–T gear 1 tooth at root. 56 lb. load barely reached
18	May 11	30	40	700	...	20	120	116	30–T gear 14 teeth out at root. 40–T gear 3 teeth out at root
19	May 11	30	40	720	...	20	120	120	30–T gear 8 teeth out at root. 40–T gear 3 teeth out at root
20	May 11	30	40	710	...	20	120	123	30–T gear 11 teeth out at root. 40–T gear 3 teeth out at root, and 1 broken
21	May 11	30	40	715	...	70	100	112	30–T gear 13 teeth stripped and 7 broken. 40–T gear 6 teeth stripped, 2 broken, 2 cracked at root
22	May 15	30	40	690	...	70	100	108	30–T gear 18 teeth stripped. 40–T gear 8 teeth stripped

TABLE 3—Continued

23	May 15	30	40	720	...	70	100	106	30-T gear 11 teeth stripped and 1 broken. 40-T gear 9 teeth stripped
24	May 15	30	40	720	...	70	70	102	30-T gear 28 teeth stripped and 1 broken. 40-T gear 15 teeth stripped
25	May 15	30	40	720	...	70	70	102	30-T gear 11 teeth stripped and 2 broken. 40-T gear 9 teeth stripped
26	May 15	30	40	720	...	70	70	103	30-T gear 16 teeth stripped and 1 broken. 40-T gear 15 teeth stripped and 1 broken
27	May 15	30	40	720	...	70	40	92	Neither gear broken. Circuit breaker out. Motor stopped
28	May 15	30	40	710	...	70	40	98	30-T gear had all teeth stripped. 40-T gear 14 teeth stripped and 1 broken
29	May 15	30	40	710	...	70	40	96	Neither gear broken. Circuit breaker out. Motor stopped
30	May 15	30	40	680	...	70	40	100	Neither gear broken. Circuit breaker out. Motor stopped
31	May 15	30	40	680	...	70	40	102	Neither gear broken. Circuit breaker out. Motor stopped. No use trying to break at this speed. Limit of motor. Same gears used in tests 29, 30 and 31; practically one run
32	May 15	30	40	720	...	100	70	108	Same gears used as in tests 29, 30 and 31. 30-T gear had 24 teeth stripped and 1 broken. 40-T gear had 19 teeth stripped
33	May 15	30	40	720	...	100	70	104	30-T gear had all teeth stripped. 40-T gear had 21 teeth stripped and 3 broken
34	May 15	30	40	720	...	100	70	106	30-T gear had all teeth stripped. 40-T gear had 19 teeth stripped. 106 lb. load barely reached
35	May 15	30	60	720	...	70	70	154	30-T gear had 16 teeth stripped. 60-T gear had 8 teeth stripped
36	May 15	30	60	720	...	70	70	156	30-T gear had 16 teeth stripped and 2 broken. 60-T gear had 5 teeth stripped, 1 broken
37	May 15	30	60	720	...	70	70	150	30-T gear had 20 teeth stripped. 60-T gear had 6 teeth stripped
38	May 16	30	80	710	...	70	70	202	Intermediate bearing slipped, allowing load to come on ends of teeth. 30-T gear stripped 6 teeth. 80-T gear stripped 2 teeth
39	May 16	30	80	725	...	70	70	204	30-T gear stripped 5 teeth. 80-T gear stripped 1 tooth; broke into 3 pieces, split at key-way

TABLE 3—Continued

Test No.	Date, 1912	Gears Tested 10 Pitch, Cast Iron		Motor r.p.m. Observed Tacho-meter	Brake r.p.m. Observed Tacho-meter	Change Gears 8 Pitch, Steel		Final Net Brake Load, Lb.	Notes made at Conclusion of Each Test
		Driver	Driven			Sprocket Shaft	Inter-mediate Shaft		
40	May 16	30	80	720	...	70	70	206	30–T gear not broken at all. No teeth broken out of 80–T gear; broken into 2 pieces, split at key-way. Key-way found enlarged and key had to be reset by calking more than usual amount
41	May 16	30	100	720	...	70	70	250	Same 30–T gear as used in test 40. Again not broken at all. No teeth broken out of 100–T gear, which broke into 4 pieces; split at key-way. Not necessary to try any more of these as they split at key-way. Try later the "German" key
42	May 16	30	30	720	...	70	70	72	Same 30–T gear as driver as used in tests 40 and 41; stripped 20 teeth and broke 1. Driven 30–T gear stripped 26 teeth
43	May 16	30	30	720	...	70	70	70	Driver 30–T gear stripped 24 teeth and broke 1. Driven 30–T gear stripped 24 teeth
44	May 16	30	30	720	...	70	70	68	Driver 30–T gear stripped 21 teeth and broke 1. Driven 30–T gear stripped 23 teeth
45	May 21	30	80	720	...	70	70	198	"German" key used. Same 30–T gear used as in test 6; one tooth broken or jambed. 80–T gear broken into 4 pieces, split at key-way. Key, obviously too high, canted. Ground off 1/32 in. of top. No teeth stripped
46	May 21	30	100	720	...	70	70	246	"German" key used. Same 30–T gear used as in test 6; not broken at all. 100–T gear broken into 6 pieces, split at key-way. No teeth stripped
47	May 21	40	40	720	...	30	40	114	Driver 40–T gear stripped all teeth. Driven 40–T gear stripped 12 teeth and broke 2. "German" keys used

TABLE 4 TESTS OF 10 DIAMETRAL PITCH, CAST IRON, 14½ DEG. INVOLUTE GEARS
COMPUTED RESULTS OF SECOND SERIES OF RUNS

SPEED CORRECTIONS MADE FOR ERROR OF TACHOMETER. BRAKE ARM = 32 IN. ZERO BRAKE LOAD = 3.5 LB., BALANCED ON UPPER SCALE-BEAM

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min.	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 p. Steel Driver	Intermediate 8 p. Steel Driven	Intermediate 10 p. Cast Iron Driver	Brake 10 p. Cast Iron Driven						
1	20	120	20	30	24.6	0.7854	68	19.32	1451	Reliable run
2	20	120	20	30	24.7	0.7854	78	19.40	1664	Reliable run
3	20	120	20	30	24.7	0.7854	76	19.40	1621	Reliable run
4	70	70	20	30	143.96	0.7854	62	113.07	1323	Reliable run
5	70	70	20	30	145.5	0.7854	58	114.28	1237	Reliable run
6	70	70	20	30	145.5	0.7854	60	114.28	1280	Reliable run. No teeth broken out of either gear
8	100	40	20	30	534.0	0.7854	52	419.40	1109	Reliable run. Tests Nos. 8–47 inclusive made at higher voltage than tests Nos. 1–7 inclusive
9	100	40	20	30	555.0	0.7854	58	435.90	1237	Reliable run
10	100	40	20	30	565.0	0.7854	58	443.75	1237	Reliable run
11	120	40	20	30	660.0	0.7854	54	518.36	1152	Reliable run
12	120	40	20	30	660.0	0.7854	60	518.36	1280	Reliable run. No teeth broken out of either gear
13	120	40	20	30	660.0	0.7854	52	518.36	1109	Reliable run
14	120	30	20	30	931.2	0.7854	48	731.36	1024	Nothing broken. Runs at this speed stopped. Too fast for brake
7	100	40	20	30	395.4	0.7854	52	310.55	1109	Nothing broken. Motor stalled. New electrical connections made
15	70	40	20	30	410.0	0.7854	52	322.01	1109	Reliable run, but load was probably a little less than 1109 lb. See Log, Table 3
16	70	40	20	30	406.0	0.7854	54	318.87	1152	Reliable run

TABLE 4—Continued

Test No.	Teeth in Gears				Correct r.p.m. Brake Gear	Pitch Circumference in Ft. Brake Gear	Net Brake Load in Lb.	Pitch Speed Brake Gear, Ft. per Min.	Equivalent Load at Teeth, in Lb.	Remarks
	Change Gears		Tested Gears							
	Sprocket 8 p. Steel Driver	Intermediate 8 p. Steel Driven	Intermediate 10 p. Cast Iron Driver	Brake 10 p. Cast Iron Driven						
17	70	40	20	30	410.0	0.7854	56	322.01	1195	Reliable run, but load was probably a little less than 1195 lb. See Log, Table 3
18	20	120	30	40	44.93	1.047	116	47.05	1856	Reliable run
19	20	120	30	40	46.24	1.047	120	48.42	1920	Reliable run
20	20	120	30	40	45.59	1.047	123	47.72	1968	Reliable run
21	70	100	30	40	192.76	1.047	112	201.86	1792	Reliable run
22	70	100	30	40	186.02	1.047	108	194.80	1728	Reliable run
23	70	100	30	40	194.78	1.047	106	203.97	1696	Reliable run
24	70	70	30	40	277.3	1.047	102	290.39	1632	Reliable run
25	70	70	30	40	277.3	.....	102	290.39	1632	Reliable run
26	70	70	30	40	277.3	.....	103	290.39	1648	Reliable run
27	70	40	30	40	.....	1.047	92	.....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current
28	70	40	30	40	478.53	1.047	98	501.12	1568	The three following runs indicate that this is probably faulty. Same gears used as in test No. 27
29	70	40	30	40	.....	1.047	96	.....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current. Same gears used
30	70	40	30	40	.....	1.047	100	.....	....	Neither gear broken. Motor stalled. Circuit breaker reset for heavier current. Same gears used
31	70	40	30	40	458.48	1.047	102	480.12	1632	Neither gear broken. Motor stalled. Abandoned attempt to run at this speed. Same gears used

TABLE 4—Continued

32	100	70	30	40	396.14	1.047	108	414.84	1728	Reliable run
33	100	70	30	40	396.14	1.047	104	414.84	1664	Reliable run
34	100	70	30	40	396.14	1.047	106	414.84	1696	Reliable run
35	70	70	30	60	184.86	1.571	154	290.38	1643	Reliable run
36	70	70	30	60	184.86	1.571	156	290.38	1664	Reliable run
37	70	70	30	60	184.86	1.571	150	290.38	1600	Reliable run
38	70	70	30	80	136.72	2.094	202	286.35	1616	Reliable run, except that bearing slipped allowing load to come on ends of teeth, hence 1616 lb. load is too low
39	70	70	30	80	139.61	2.094	204	292.40	1632	Reliable run
40	70	70	30	80	138.65	2.094	206	290.39	1648	Reliable run, but no teeth were stripped
45	70	70	30	80	138.65	2.094	198	290.39	1584	Unreliable run. Gear split. No teeth stripped. "German" key; see Log, Table 3
41	70	70	30	100	110.92	2.618	250	290.39	1600	Unreliable run. Gear split. No teeth broken. See log, Table 3
46	70	70	30	100	110.92	2.618	246	290.39	1574	Unreliable run. Gear split. No teeth broken. See Log, Table 3, "German" key
42	70	70	30	30	369.73	0.7854	72	290.39	1536	Reliable run
43	70	70	30	30	369.73	0.7854	70	290.39	1493	Reliable run
44	70	70	30	30	369.73	0.7854	68	290.39	1451	Reliable run
47	30	40	40	40	277.30	1.047	114	290.33	1824	Reliable run

of the equivalent breaking loads was taken for each set, and the curve drawn through these average points as near as might be.

14 In Fig. 4 the curve was extended by inference to the zero velocity line. It will be noted that this breaking load curve falls off with increase of speed up to a pitch velocity of something less than 300 ft. per min. and then apparently starts to

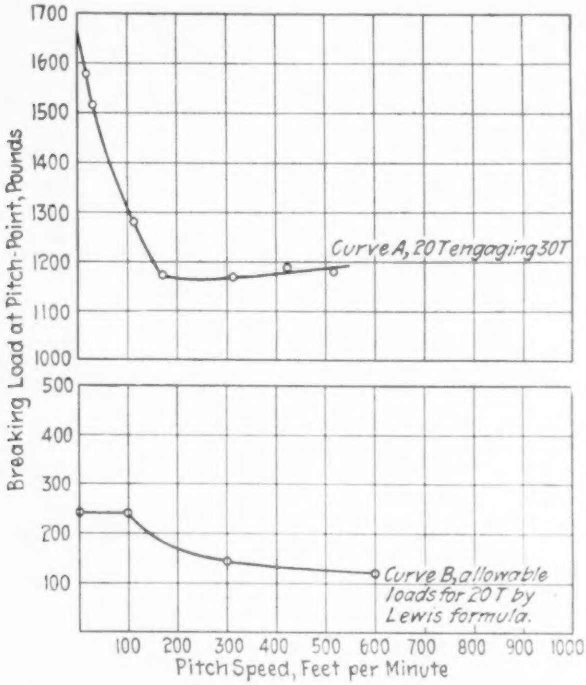


Fig. 4 Results of Tests with 20-Tooth Pinions

rise again as though the maximum percussive effect had been passed. This fact, and the further one that within the limits of the actual tests the range of average breaking load was only from 1579 lb. at 19.4 ft. per min. down to 1169 lb. at 312.7 ft. pitch velocity, about 25 per cent, are two of the most striking points appearing in this investigation.

15 These phenomena might be questioned on the score that all of the 20-tooth gears but one broke entirely apart as well as having teeth stripped. But Fig. 5 shows the same general form for all tests involving 30- and 40-tooth gears in mesh;

and in this case there can be no question of anything but the strength of the teeth themselves, as it was only at the teeth that any of these gears broke.

16 It is an unfortunate fact that the limit of the motor's power was such as to make it impossible to rupture gears at higher speeds, thus enabling us to follow the curve and observe whether the apparent rise in breaking strength continued with increase in pitch speed. No attempt has been made to make this a smooth curve; the points corresponding to the averages obtained by experiment have merely been joined.

17 It is clear, however, even from these somewhat limited

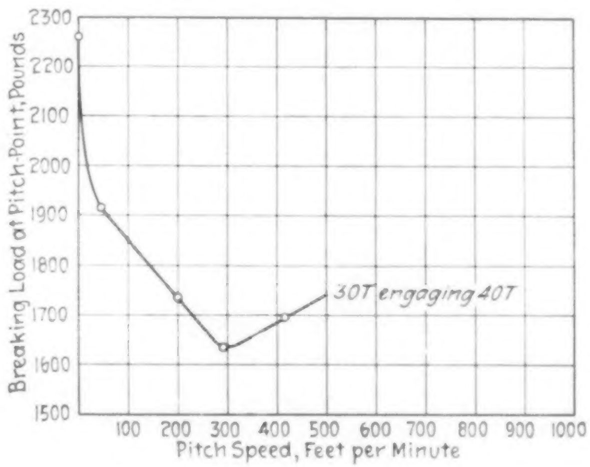


FIG. 5 RESULTS OF TESTS WITH 30 AND 40-TOOTH GEARS

experiments, that the impact or percussive effect with increase of speed is not nearly so great, with these modern, accurately cut gears, as has commonly been supposed. In both cases the curves show that the minimum breaking strength (at a pitch speed of about 300 ft. per minute) is more than seven-eighths that at a pitch speed of about 100 ft. per min. and more than seven-tenths that at zero pitch speed. And it is not theoretically impossible that the breaking load actually rises with increase of speed, after a certain critical speed has been passed, until it would approximate the static breaking load if it were not for the increasing tendency of the teeth to tear loose due to centri-

fugal force. At an infinite speed the repetitive stress would become a continuous one. As Curve *A*, Fig. 4 shows the actual breaking strength of 20 teeth, 10 pitch, cast-iron gears at different speeds, it is interesting to compare with this the allowable loads for such gears computed for the same speeds by the Lewis formula (see Appendix No. 1). Curve *B*, expressing these, is shown at the bottom of Fig. 4. The discrepancy between the two curves is obvious. They do not even have the same general form.

18 From Par. 4 it is seen that Mr. Lewis's formulae and tables were intended to provide a factor of safety of 3 for pitch speeds of 100 ft. per min. or less, of 5 for 300 ft., and 6 for 600 ft., the differences being intended to cover presumed increases in impact effect. Using Mr. Lewis's accepted formula for 15-deg. involute teeth as being applicable to the Brown & Sharpe standard $14\frac{1}{2}$-deg. involute teeth, namely,

$$W = s\ p\ f\ 0.124 - \frac{0.684}{n}$$

W = working strength of tooth
s = working stress of material per sq. in.
p = pitch of teeth in inches
f = face of teeth in inches
n = number of teeth,

Table 5 has been prepared to show the relation between the computed working strength and the actual breaking loads for the 20 and 30 tooth gears within the range of these experiments as shown by the plotted curves on their legitimate inferential extension.

19 When the actual breaking loads under running conditions have been determined, as they have in the above cases, the results provide the allowances for pitch-speed variations and it is difficult to see any reason for not using a uniform factor of safety for conditions which are uniform in other regards than pitch-speeds, to provide for the possibilities of faulty material, poor alignment, sudden applications or reversals of stress, and possible overloads. Just how great this factor need be is a question to be settled by the designer in each individual case. But in any case it would be a true factor of safety and not a factor of ignorance.

20 Mr. Lewis's Table II may be looked upon as a table of

TABLE 5 RELATION BETWEEN COMPUTED WORKING STRENGTH AND BREAKING LOADS 20- AND 30-TOOTH GEARS

20T In Mesh With 30T	Pitch	Speed	Ft. per min.	
	0	100	300	600
Computed Working Load, Lewis's Formula	240	240	144	120
Actual Breaking Load, Tests	1660	1310	1165[1]	1165[1]
Corresponding Factor of Safety	6.9	5.5	8.1	9.7
Working Load, Factor of Safety = 4	413	340	290	290
30T In Mesh With 40T				
Computed Working Load, Lewis's Formula	270	270	163.2	135
Actual Breaking Load, Tests	2260	1850	1640	1640[1]
Corresponding Factor of Safety	8.4	6.9	10	12.2
Working Factor of Safety = 4	565	463	410	410

[1] Neglecting apparent rise in curve.

TABLE 6.—VELOCITY COEFFICIENTS

Pitch Velocity, Ft. per Min.	000	100	150	200	300	400	500
Ratio $\frac{\text{Breaking Load at Given Vel.}}{\text{Breaking Load at Zero Vel.}}$ Data from Fig. 4	$\frac{1660}{1660}=1.0$	$\frac{1310}{1660}=0.789$	$\frac{1205}{1660}=0.725$	$\frac{1165}{1660}=0.702$	$\frac{1165}{1660}=0.702$	$\frac{1175}{1660}=0.707$	$\frac{1185}{1660}=0.712$
Ratio $\frac{\text{Breaking Load at Given Vel.}}{\text{Breaking Load at Zero Vel.}}$ Data from Fig. 5	$\frac{2260}{2260}=1.0$	$\frac{1850}{2260}=0.819$	$\frac{1790}{2260}=0.792$	$\frac{1735}{2260}=0.768$	$\frac{1640}{2260}=0.726$	$\frac{1680}{2260}=0.743$	$\frac{1740}{2260}=0.770$
v—Velocity Coefficient, Safe	1.00	0.80	0.75	0.72	0.70	0.68	0.66

factors representing a combination of the modulus of rupture, a factor of safety, and a coefficient depending upon velocity. It would seem better to separate these. The value of the modulus of rupture is discussed in Par. 24. For values to be taken for the factor of safety the writer agrees with Mr. Christie as quoted in Par. 4.

21 The values of the coefficient to provide for the effect of pitch-velocity v can be determined from the results of the tests shown in Fig. 4, Curve A and Fig. 5. These results are expressed in Table 6, together with a series of values of v based upon them and so selected as to lean toward the side of extra safety.

22 To check the material of the gears used in these tests and to determine whether it was normal or of exceptional strength, 24 test specimens were cut from 12 of the 30-tooth gears, selected at random, as shown in Fig. 6. These specimens were about ¼ in. thick, 11/16 in. wide and 2¼ in. long. Seventeen similar specimens were cut from castings made in the university's foundry and known to be of ordinary composition and quality. The specimens were all tested in flexure with the load aplied at the middle of a span of 1¾ in. The data and results of these tests are given in full in Table 7. The modulus of rupture of the gear material test pieces was 38,737 lb., while that of the check specimens was 39,049 lb. It is legitimate to conclude that the material of the gears used in these tests, while of first-rate quality, was not unusual or exceptional. The results of the tests may therefore be accepted as representing typical material. In general a value of 36,000 may be reasonably used for the modulus of rupture for cast-iron gear teeth.

23 The Lewis formula is based upon the theory of flexure under the assumptions that the entire load is borne by a single tooth, the point of application of the load being the extreme end of the tooth, and the direction of the acting force being normal to the tooth profile at that point. The line of action of the force is considered carried to the center line of the tooth (Fig. 7) with W equal to the transverse component as applied at this point. Subsequently it treats W as the load at the pitch circumference. The effects of the compressive and shear components are ignored. Fracture is assumed to take place on a plane section, the position of which is determined by inscribing a parabola of uniform

strength with the vertex lying at the intersection of the center line of the tooth and the line of action of the force. The dangerous section is the plane section containing the points of tangency of the parabola with the tooth outline. That these assumptions are too severe or unfavorable for accurately cut gears is evident from the result of these tests, as will be shown later.

24 To check the applicability of the Lewis formula to the

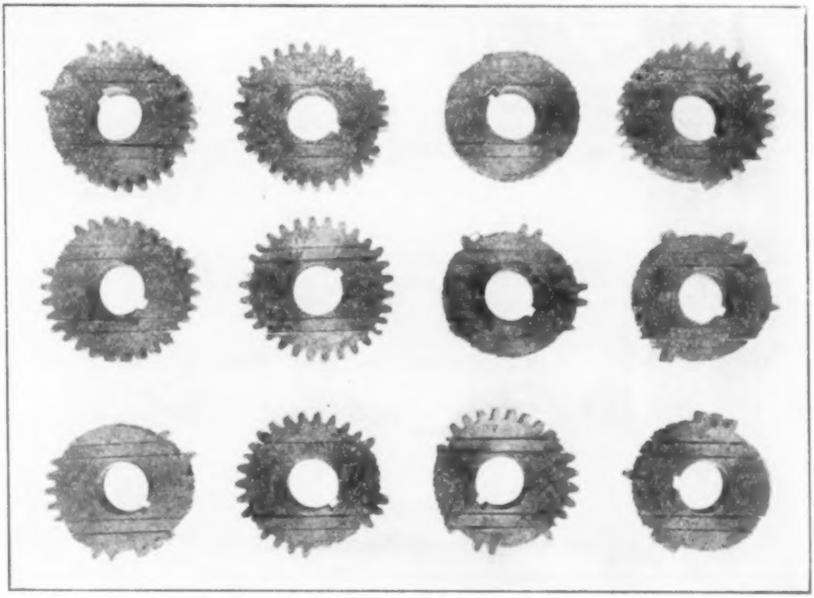


FIG. 6 METHOD OF OBTAINING TEST SPECIMENS

form of gear tooth employed in these experiments, the teeth of the 20, 30, 40, 80, 100 and 150 tooth 14½-deg. involute gears were carefully measured by gear-tooth micrometer calipers. They then, together with a theoretical rack tooth of the same system, were accurately drawn to a scale of 100 times full size. In these teeth, parabolas were inscribed by the Lewis method and the dangerous section located. The data and results are given in Appendix No. 2. The factors for strength do not seriously vary from those given by the Lewis expression[1] for 15-deg involute teeth, i.e., $0.124 - \frac{0.684}{n}$. If the assumptions of Par. 23 are

[1]American Machinist, June 22, 1893.

TABLE 7 TESTS OF MATERIALS, DATA AND RESULTS, JUNE 26, 1912

SPECIMENS CUT FROM GEARS

Mark	Thickness	Width	Breaking Load at Mid-point	Corresponding Modulus of Rupture	Average Modulus of Rupture	Remarks.
05 *A*	0.251	1.066	980	38305	....	
05*B*	0.251	1.065	975	38145	38225	
08*A*	0.251	1.065	1000	39123	....	
08*B*	0.251	1.063	1000	39196	39160	Small blow-hole on top.
09*A*	0.248	1.066	1167.5	46343	....	
09*B*	0.250	1.065	1197.5	47225	46784	
1*A*	0.251	1.063	917.5	35969	....	Small blow-hole on bottom.
1*A*	0.251	1.063	967.5	37922	36946	
17*A*	0.248	1.060	1106	44533	....	
17*B*	0.250	1.058	1156.5	45910	45222	
19*A*	0.245	1.060	950	39194	....	
19*B*	0.248	1.061	970	39020	39107	
28*A*	0.251	1.067	....	....	....	Flawed. No reading.
28*B*	0.251	1.067	850	33192	33192	
32*A*	0.251	1.060	1000	39307	....	
32*B*	0.250	1.061	995	39387	39347	
37*A*	0.251	1.06	895	35081	....	
37*B*	0.251	1.063	904	35434	35258	
39*A*	0.251	1.0635	842.5	33008	....	
39*B*	0.250	1.0635	870	34358	33683	
43.*A*	0.251	1.060	867.5	34099	....	
43.*B*	0.251	1.060	874	34355	34227	
43..*A*	0.251	1.056	1087.5	42909	....	
43..*B*	0.251	1.054	1125	44472	43691	
				Grand Aver.	38737	

CHECK SPECIMENS, ORDINARY CAST IRON

A	0.251	1.0615	1000	39252	....	
B	0.251	1.063	....	....	....	Broke before initial load,900lb
C	0.250	1.058	990	39300	....	
D	0.252	1.062	962.5	37463	....	
E	0.252	1.062	975	37862	....	
F	0.251	1.059	1000	39354	....	
G	0.252	1.062	920	35809	....	
H	0.248	1.059	990	39899	....	
I	0.257	1.062	1195	44720	....	
J	0.254	1.058	880	33842	....	
K	0.251	1.057	1018	40129	....	
L	0.247	1.061	842	34145	....	
M	0.245	1.064	....	....	....	Void. Flawed.
N	0.250	1.0625	1014	40083	....	
O	0.250	1.0625	1136	44905	....	
P	0.252	1.058	945	36930	....	
Q	0.252	1.062	1080	42037	....	
				Grand Aver.	39049	

Distance between supports = 1.75 in. in all cases.

correct, the Lewis formula should give the actual static breaking loads when the actual modulus of rupture is inserted for s.

25 To test this, consider first the case of a 20-tooth 10-pitch, 1 1/16 in. face, cast-iron pinion, the material having a modulus of rupture of 39,000. Substitution in the Lewis formula

$$W = \text{spf}\ (0.124 - \frac{0.684)}{n}$$

gives

$$W = 39{,}000 \times 0.31416 \times 1.0625 \left(0.124 - \frac{0.684}{20}\right)$$

therefore

$$W = 1170 \text{ lb.}$$

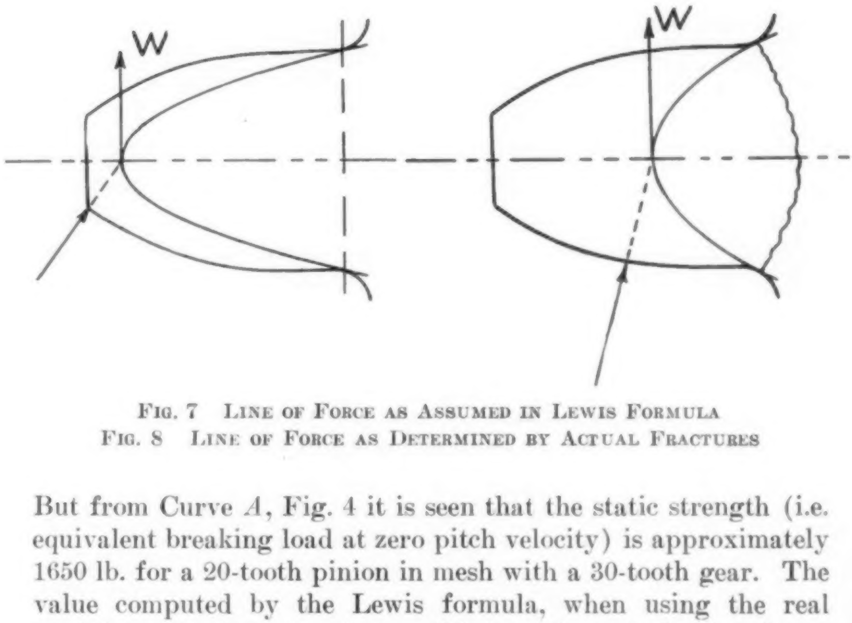


FIG. 7 LINE OF FORCE AS ASSUMED IN LEWIS FORMULA
FIG. 8 LINE OF FORCE AS DETERMINED BY ACTUAL FRACTURES

But from Curve A, Fig. 4 it is seen that the static strength (i.e. equivalent breaking load at zero pitch velocity) is approximately 1650 lb. for a 20-tooth pinion in mesh with a 30-tooth gear. The value computed by the Lewis formula, when using the real modulus of rupture, is only two-thirds of the actual value shown by the tests. Similarly, the Lewis formula gives a value of 1317 lb. for the static breaking strength of the 30-tooth gear, while Fig. 5 shows this to be, for a 30-tooth meshing with a 40-tooth, 2260 lb. Again the Lewis value is much too low, being but 60 per cent of the actual test result.

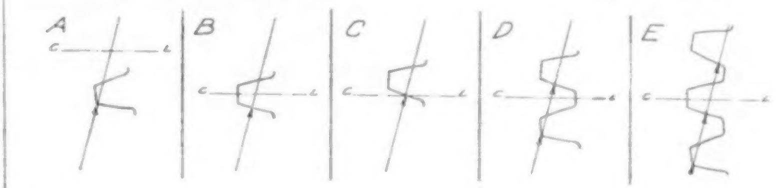
26 These discrepancies demonstrate that the assumptions

upon which the Lewis formula are based are too severe and they indicate the necessity of taking into consideration the length of the arc of action; for if the length of the arc of action exceeds the pitch arc, it is evident that the previously engaging teeth will not have gone out of action at the time when the load first comes on the newly engaging teeth, and it is only at the instant of engagement or disengagement that the load can be applied at the end of a tooth. This is not only theoretically so but it can be counted on with reasonable certainty for modern, accurately cut gears. It must be borne in mind that this is the least stiff position of the tooth and that it would therefore tend to yield slightly under the load thus allowing another pair of teeth to come into engagement or remain in engagement. This elastic yielding would be apt to make up for the slight errors in tooth spacing. The mathematical discussion of the arc of action and its numerical determination for the gear combinations used in these experiments will be found in Appendix No. 3.

27 In order to test the actual static strength of the teeth in various positions, and with one, two or if possible, three pairs engaging simultaneously, experiments were carried out with results as shown in Table 8. The loads were applied in each case by a steel 30-tooth pinion with all teeth save three single ones, two pairs of two, and one set of three having been milled off in such a way as to leave gaps between the remaining teeth. Because of this there could be positive assurance that only as many pairs of teeth engaged in any test as was intended. The gears were set up with the teeth engaging in the position desired, the cast-iron gear to be tested being the one on the brake shaft and a surface gage being used to test the accuracy of conditions. All play was taken up and the brake was tightened down with emery cloth between it and the wheel so as to prevent all slip. The breaking power was supplied by means of a special hand-operated lever instead of by motor.

28 Position A corresponds to the loading assumptions of the Lewis formula and it is interesting to compare these results with those obtained by using the Lewis formula with a modulus of rupture of 39,000 lb. Table 9 presents this comparison in concise form.

TABLE 8. EQUIVALENT STATIC BREAKING LOADS W AT PITCH LINE—SERIES 4, SEPTEMBER 11-13, 1912



10-Pitch, Cast-Iron Gear	Position of Stressed Teeth and Equivalent Breaking Load									
	Single Tooth Loaded						2-Teeth Loaded		3-[?] Teeth Loaded	
	A		B		C		D		E	
	Test No.	W	Test No.	W	Test No.	W	Test No.	W	Test No.	W
20-T	4	1216	12	1440	1	1632	7	2112	9	1600
	5	1184	13	1476	2	1696	8	2176	10	1472
	6	1152	..	....	3	1664	..	....	11	1632
	Aver.	1184	Aver.	1458	Aver.	1664	Aver.	2144	Aver.	1568
30-T	18	1472	25	1877	14	1941	20	2603	23	1835
	19	1429	26	1707	15-16	Void	21	2176	24	1856
	..	....	27	1728	17	1899	22	2965	..	....
	Aver.	1451	Aver.	1771	Aver.	1920	Aver.	2581	Aver.	1846
40-T	31	Void	39	2416	28	2416	34	3360	36	Void
	32	1616	40	2336	29	2272	35	3360	37	2560
	33	1600	..	....	30	2208	..	....	38	2528
	Aver.	1608	Aver.	2376	Aver.	2299	Aver.	3360	Aver.	2544

TABLE 9 EQUIVALENT BREAKING LOADS

Size of Gear	20 T	30 T	40 T
W by Test, Position A	1184	1451	1608
W computed by Lewis Formula	1170	1317	1392

29 The test values of W, as given in Table 9, correspond to a formula for this position of engagement and static conditions:

$$W = \text{spf } \left(0.154 - \frac{1.26}{n}\right)$$

The derivation of this formula from the experimental results will be found in Appendix No. 4. This equation would make the static strength of single teeth vary from that of a theoretically correct rack tooth down to a 20-tooth pinion in the ratio of 0.154 to 0.091; in other words, the rack tooth would have 1.69 times the strength of the 20-tooth pinion. The Lewis formula would give a ratio of 0.124 to 0.090 for the same gears. The discrepancy can be explained by the fact that the theory of flexure is recognized as only approximately correct, particularly for oblique forces, material which has not the same strength in tension as in compression, and for stresses beyond the elastic limit. Furthermore the Lewis factors neglect entirely the effect

of the angularity of the force in producing a compressive stress. In the case of the 20-tooth gear this angularity is such as to make the compressive component of the acting force more than 50 per cent of the transverse component. The theory of flexure upon which the Lewis formula is based would have the fracture occur on a plane section (as shown in Fig. 7) and would have the breaking loads vary, approximately as the square of the thickness of the tooth at this section. As a matter of fact the fracture of the teeth in every case took place lower down on the root than at the points of tangency of parabolas inscribed on the Lewis assumption. The fractures took place rather at the points of tangency of parabolas constructed for loads applied at the pitch points as shown in Fig. 8, and the breaking loads varied more nearly in proportion to the squares of the thickness at this section. The surface of the break invariably was in the form of a curve running down into the root as shown and in no case did a tooth break square across.

30 Mention is made in Par. 26 of the necessity of taking the arc of action into consideration in dealing with the load-carrying strength of gear teeth. It would seem that the breaking load of a gear would depend not solely upon itself but also upon the gear with which it meshed. The larger the meshing gear, the greater the arc of action; and the greater the arc of action, the more likely the load is to be borne on two or more pairs of teeth simultaneously and the less likely that the entire load will fall on a single tooth in its position of least strength. It is desirable to see, then, what effect, if any, the length of arc of action has on the breaking load. To see what the effect was under static (zero pitch velocity) conditions a series of tests (Series 5) was conducted. Tests were made of 30-tooth gears in mesh with 30, 40, 60 and 80 tooth gears. It was found in making these tests that the position of the teeth in engagement made a vast difference in results. It was therefore necessary to find by trial the position of engagement corresponding to the least carrying power of the tooth. The results are given in Table 10. There is nothing to account for the foregoing variation in strength except the differences in arc of action. It was the intention to carry similar tests out on 30-tooth meshing with 80-tooth and 100-tooth, but it was found in trying the 30 and 80 pair,

TABLE 10 STATIC STRENGTH. SERIES 5, SEPTEMBER 26-27, 1912

Breaking Loads W of 30T in mesh with

30T			40T			60T		
Series	Test	W	Series	Test	W	Series	Test	W
4	23	1835	5	1	2480	5	14	2453
4	24	1856	5	6	2368	5	15	2453
5	17	2475	5	7	2080			
5	18	2304	5	8	2112			
Average.............2118 Arc of Action, In...0.61158			Average.............2260 Arc of Action, In...0.62814			Average............2453 Arc of Action, In...0.64919		

that the rim of the 80-tooth broke, cracking the arms and hub, before a breaking load of the teeth could be reached. Fig. 9, Curve A, shows the results of these tests, the ordinates being equivalent breaking loads at pitch point, and the abscissae being arcs of contact. There being no unbroken 20-tooth gears to use in Series 5, the value for static strength of 30-tooth meshing with 20-teeth (arc of action, 0.53938 in.) is taken from Fig. 4 Curve A.

31 From Table 9 it is seen that the static strength of a single tooth of a 30-tooth gear in its weakest position = 1451 lb. This would measure the strength of teeth with an arc of action of 0.314 in. or less, i.e. when the $\frac{\text{arc of action}}{\text{pitch arc}} \leqq 1$. From Fig. 9, Curve A, for a 30-tooth meshing with a 20-tooth the arc of action equals 0.53938 in. and the corresponding $W = 1660$ lb. From these facts and Table 10 it is possible to construct Table 11 to show the relationship, under static conditions, existing between length of arc of action and equivalent breaking load.

TABLE 11 RELATION BETWEEN ARC OF ACTION AND EQUIVALENT STATIC LOAD

Ratio $\frac{\text{Arc of Action}}{\text{Pitch Arc......}}$	$\frac{0.31416}{0.31416}=1$	$\frac{0.53938}{0.31416}=1.72$	$\frac{0.61158}{0.31416}=1.95$	$\frac{0.62814}{0.31416}=2$	$\frac{0.64919}{0.31416}=2.07$
Breaking Load.......	1451	1660	2118	2260	2453
Ratio $\frac{\text{Breaking Load}}{1451}$	1	1.14	1.46	1.56	1.69

The range in breaking strength due to varying arcs of action, from 1 to 1.69 shown by this table, is as great as that due to *form* between 20 teeth and 150 teeth (see Par. 28); and it is greater than that due to variations in velocity from zero pitch speed to over 500 ft. per min. (see Par. 21). If form of tooth and velocity are of sufficient importance to be taken into account in a formula for gear tooth strength it is demonstrated that there should also be introduced a factor for arc of action.

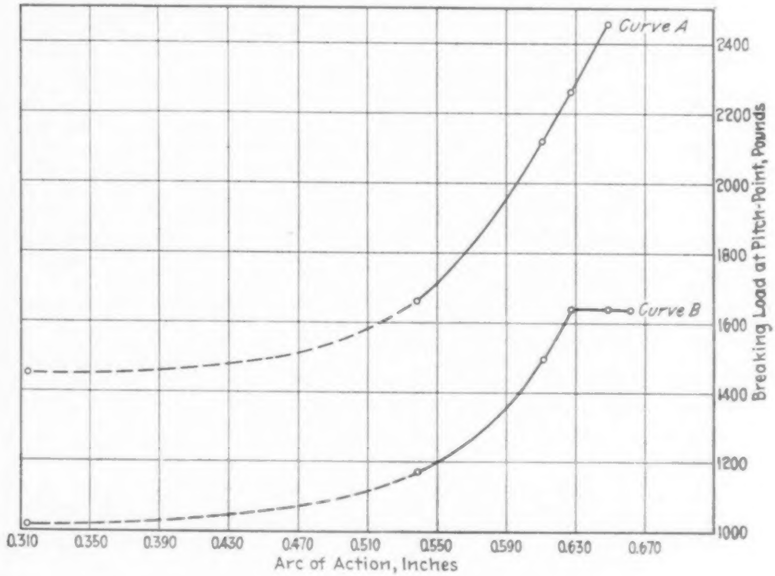


Fig. 9 Relation Between Equivalent Breaking Load and Arc of Contact

32 Further light is thrown upon the extent of the influence of the length of arc of action if from Series 1 and 2 the tests involving 30 tooth gears made at nearly a uniform pitch velocity are selected. The following table gives the results of such tests made at about 290 ft. pitch velocity. Such a selection makes the arc of action the only variable factor. The values of Table 12 are shown graphically in Fig. 9, Curve *B*. There are no running experiments involving an arc of action of 0.31416. But from Figs. 4 and 5 it is clear that the breaking strength at 300 ft. per min. is about 70 per cent of the static strength. It is therefore not improper to assume that the break-

TABLE 12 RELATION BETWEEN ARC OF CONTACT AND BREAKING LOAD

Run	Speed	Gears	Arc of Contact	Breaking Load
6	290.03	30-20	0.53938	1280
7	310.55	30-20	0.53938	1109
15	322.01	30-20	0.53938	1109
16	318.87	30-20	0.53938	1152
17	322.01	30-20	0.53938	1195
				Average....1169
42	290.39	30-30	0.61158	1536
43	290.39	30-30	0.61158	1493
44	290.39	30-30	0.61158	1451
				Average.....1493
24	290.39	30-40	0.62814	1632
25	290.39	30-40	0.62814	1632
26	290.39	30-40	0.62814	1648
				Average....1637
35	290.38	30-60	0.64919	1643
36	290.38	30-60	0.64919	1664
37	290.38	30-60	0.64919	1600
				Average....1636
38	286.35	30-80	0.66209	1616[1]
39	292.40	30-80	0.66209	1632
40	290.39	30-80	0.66209	1648[1]
				Average....1632

[1] Data indicate that these values were lower than real strength of teeth. See Table 3.

ing strength for an arc of contract of 0.31416, at a pitch velocity of 300 ft. per min. would be 70 per cent of 1451 lb., or 1016 lb. On this assumption and the data of Table 11, Table 13 has been computed.

It is interesting to note how closely this agrees with Table 9.

33 The results in Tables 11 and 13 may now be combined and a table of coefficients a constructed by which the effect of the arc of action may be introduced into the formula for the strength.

TABLE 13 RELATION BETWEEN ARC OF ACTION AND BREAKING LOAD UNDER RUNNING CONDITIONS

Ratio: $\frac{\text{Arc of Action}}{\text{Pitch Arc}}$	$\frac{0.31416}{0.31416}=1$	$\frac{0.53935}{0.31416}=1.72$	$\frac{0.61158}{0.31416}=1.95$	$\frac{0.62814}{0.31416}=2$	$\frac{0.64919}{0.31416}=2.07$	$\frac{0.66209}{0.31416}=2.11$
Breaking Load	1016	1169	1493	1637	1636	1632
Ratio: $\frac{\text{Breaking Load}}{1016}$	1	1.15	1.47	1.61	1.61	1.61

TABLE 14 COEFFICIENTS BASED ON ARC OF ACTION

Ratio: $\frac{\text{Arc of Action}}{\text{Pitch Arc}}$	1	1.4	1.6	1.7	1.8	1.9	1.95	2.00
Corresponding a	1	1.05	1.1	1.15	1.24	1.38	1.47	1.60

34 The entire formula as derived from these experiments for the safe equivalent load at pitch line may now be written

$$W=\frac{spf}{k}\left(0.154-\frac{1.26}{n}\right)va.$$

in which

W = safe working load at pitch line in pounds

s = modulus of rupture = 39,000 in these tests but ordinarily to be taken = 36,000.

p = circular pitch in inches

f = width of face in inches

k = factor of safety

n = number of teeth in gear

v = velocity coefficient from Table 6.

a = arc of action coefficient from Table 14.

35 The general effect of arcs of action and of pitch speed on the breaking strength of gear teeth as found in these experiments can best be seen by combining Fig. 4, Curve A, Fig. 5 and

Fig. 9, Curves A and B into a single isometric chart. This is done in Fig. 10. The ordinates, as heretofore, represent equiva-

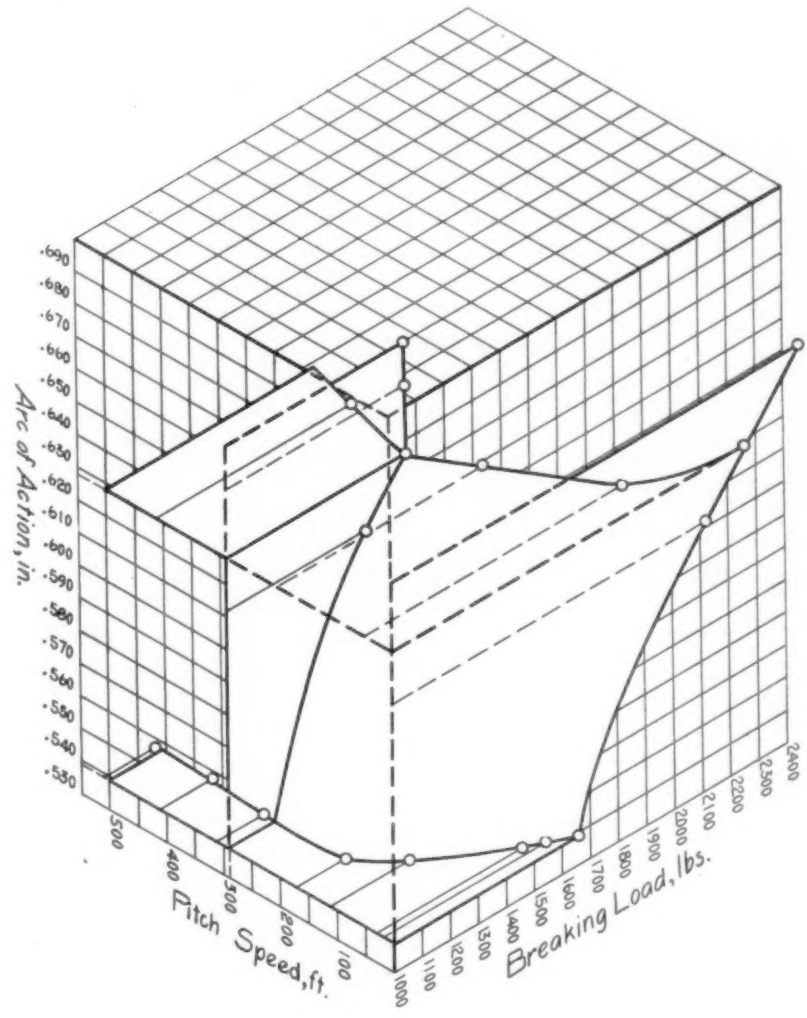


Fig. 10 Relations Between Arc of Action, Pitch Speed and Breaking Load Shown Isometrically

lent breaking loads at pitch point in pounds. One set of abscissae represents pitch speeds in feet per minute; the other set represents lengths of arc of action in inches.

36 To complete the record of this experimentation mention should be made of the tests which were designated as Series 3. In planning these it was thought that the remaining unbroken gears could have their width of face turned down to one-half its previous value, and then, the teeth being theoretically just half as strong as those of the full-width gears, tests might be conducted at higher pitch speeds and the gears ruptured within the capacity of the motor. The observed breaking load could then be doubled as the breaking load of the full-width gears and points so obtained be used to extend the curve of direct observations. The results of a very few tests proved that this could not be done. The data and results are briefly tabulated in Table 15.

TABLE 15 SERIES 3, JULY 1, 1912

Test No.	Width of Face	Change Gears		Test Gears		Motor R.P.M.	Net Brake Load, Lb.	Pitch Speed Brake Gear Ft. per Min.	Equivalent Load at Teeth, Lb.
		Sprocket	Intermediate	Intermediate	Brake				
48	17/32	70	40	30	30	720	28	508.18	597
49	1 1/16	70	40	30	30	720	66	508.18	1408
50	17/32	40	20	30	30	720	28	580.77	597
51	1 1/16	40	20	30	30	720	62	580.77	Could not break. Circuit-Breaker out.
52	1 1/16	70	40	30	30	720	66	508.18	1408 Same Gears as Test 51

It is evident from Tests 48-49 and 50-51 that the half width teeth have not one-half the strength of the full-width teeth. Only speculations can be offered as to this apparent discrepancy. It may be that the wider teeth forced better alignment conditions without unduly stressing the teeth. Or it may be that the additional mass plays some part in shock absorption. But, since observations could not be used to supplement those on full-width gears no further runs were made.

37 It would appear that the factors of safety in common use,

in so far as breaking strength of the teeth is concerned, while they might properly serve for rough cast gears, are much too large for modern, accurately cut gears; and that Mr. Lewis and Mr. Christie were quite right in assuming that the factor of safety at any speed need not be more than double that which will suffice for the slowest speed.

38 A few words may be added about the general design of the gears. The almost invariable breaking of the 20-tooth gears, 1 1/16 in. bore, 1/4 in. keyway, tends to indicate that this is just about the maximum limit of bore which can be used with these gears while getting the maximum out of the tooth strength. Any larger bore, it seems, would simply allow these gears to split at the keyway without stripping any teeth. Of the 30- and 40-tooth gears, and of the proportions of the 60-tooth there can be no criticism. In all cases the failure came at the teeth, although it is indicated that the 60-tooth gears could have borne heavier loads if they had been solid. As a whole the tests involving gears with more than 60 teeth were indecisive so far as throwing any light on the strength of the teeth was concerned. These gears were not solid nor webbed but had arms, and their less rigid design allowed them to yield and fracture at rim, arms and hub before the breaking strength of the teeth could be reached. To bring the rest of the gear up to the strength of the teeth would seem to require a larger diameter of hub, thicker rims and heavier arms. In loading the gears up to the breaking strength of the teeth considerable difficulty was encountered with shearing and battering of keys. This was practically overcome by the use of hardened tool-steel keys. A photographic log follows which shows the manner in which the gears failed.

ACKNOWLEDGMENT

The writer wishes to acknowledge his appreciation of the many kindnesses of his colleagues and the laboratory assistants. Particular thanks are due to Mr. L. E. Cutter, instructor in mechanical engineering, without whose unfailing assistance the tests could scarcely have been completed. All observations were made by the writer and Mr. Cutter personally.

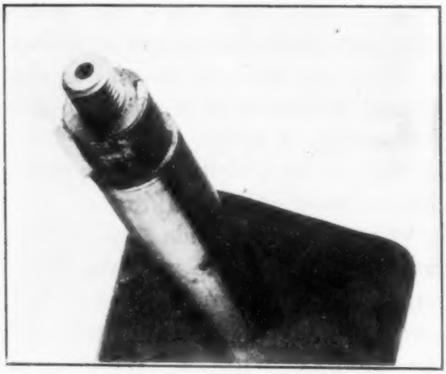


Fig. 11 Key Sheared in Test 01, Series 1

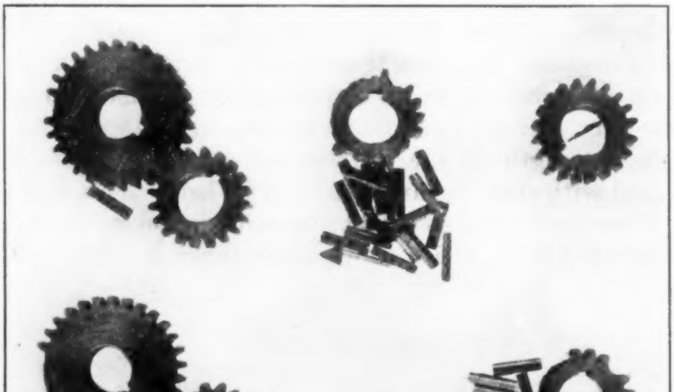


Fig. 12 20 Tooth engaging 30 Tooth, Tests 04—08 inclusive, Series 1

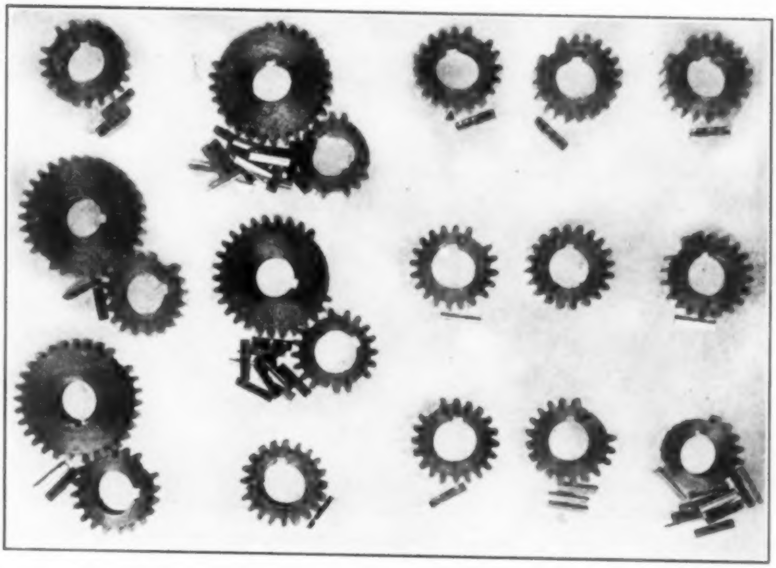


FIG. 13 20 TOOTH ENGAGING 30 TOOTH, TESTS 1—17 INCLUSIVE, SERIES 2

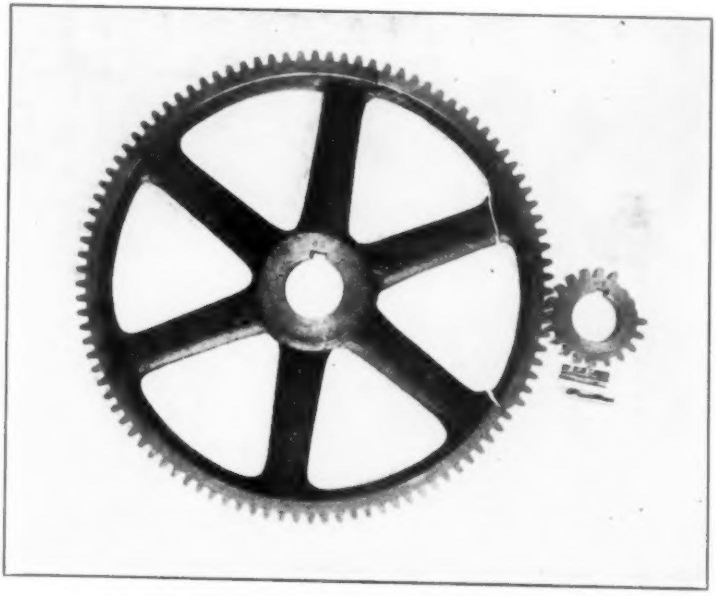


FIG. 14 20 TOOTH ENGAGING 100 TOOTH, TEST 03, SERIES 1

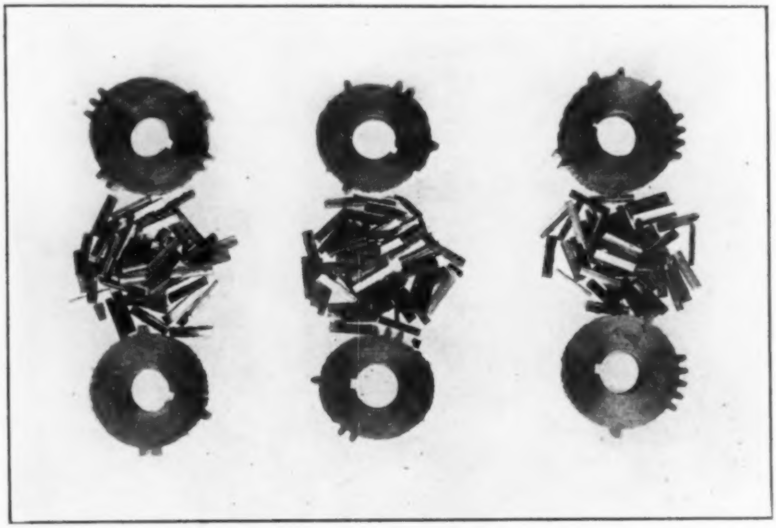


FIG. 15 30 TOOTH ENGAGING 30 TOOTH, TESTS 42—44 INCLUSIVE, SERIES 2

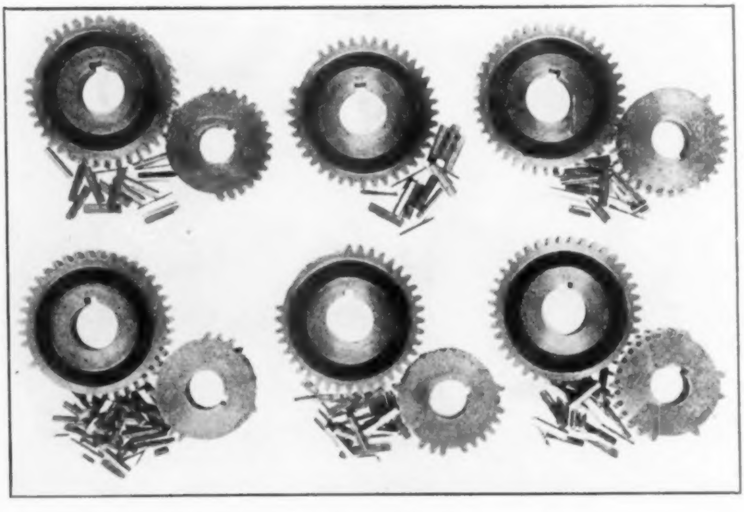


FIG. 16 30 TOOTH ENGAGING 40 TOOTH, TESTS 18—23 INCLUSIVE, SERIES 2

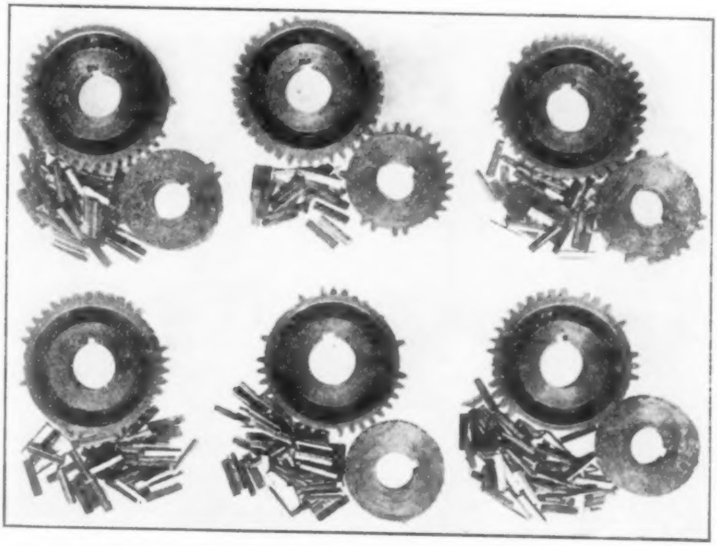


FIG. 17 30 TOOTH ENGAGING 40 TOOTH, TESTS 24—26, 32—34 INCLUSIVE, SERIES 2

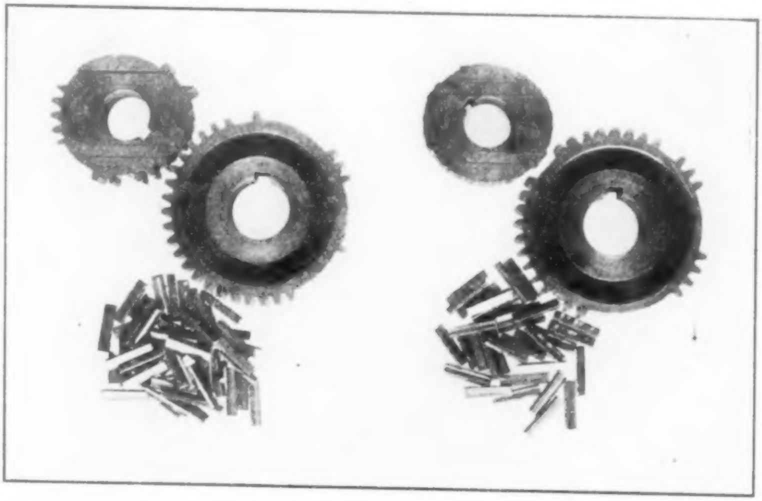


FIG. 18 30 TOOTH ENGAGING 40 TOOTH, TEST 09, SERIES 1 AND TEST 28, SERIES 2

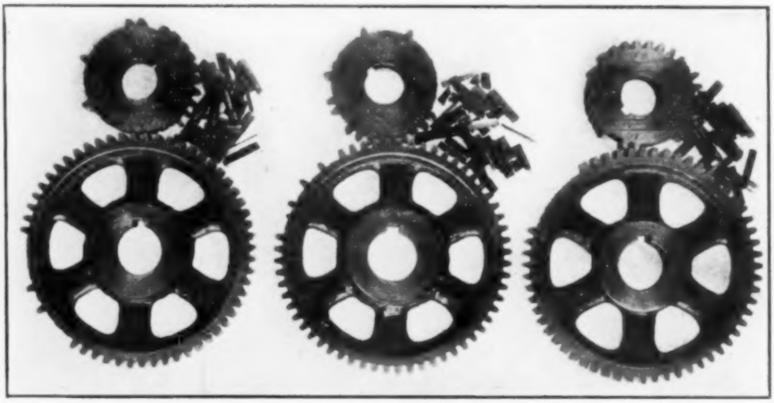


FIG. 19 30 TOOTH ENGAGING 60 TOOTH, TESTS 35—37 INCLUSIVE, SERIES 2

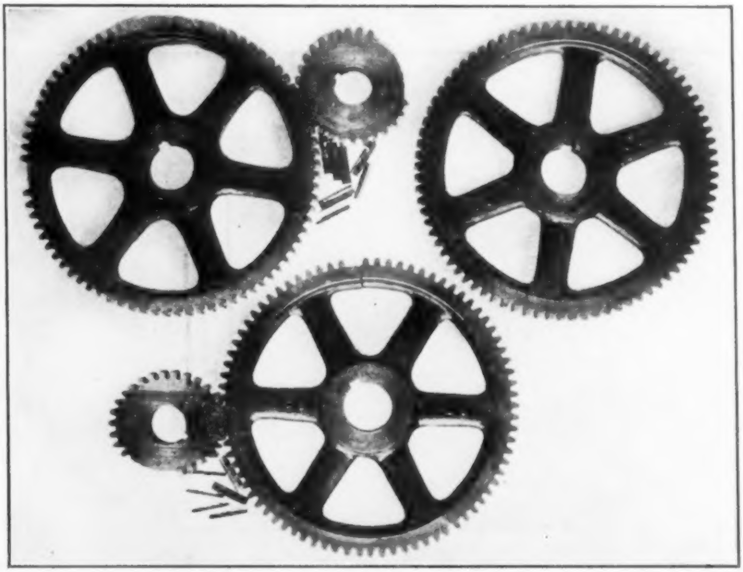


FIG. 20 30 TOOTH ENGAGING 80 TOOTH, TESTS 38—40 INCLUSIVE, SERIES 2

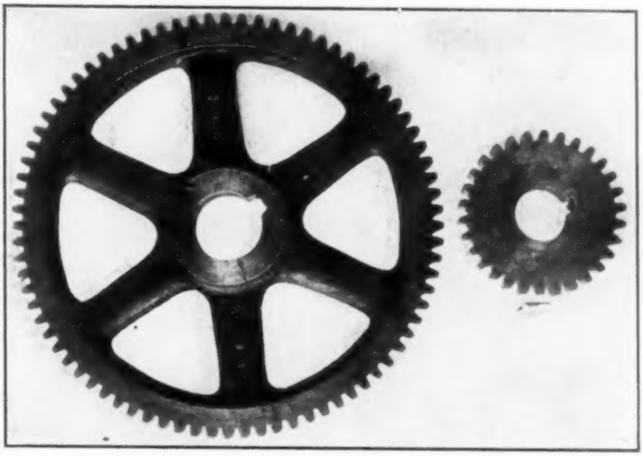


FIG. 21 30 TOOTH ENGAGING 80 TOOTH, TEST 45, SERIES 2. GERMAN HEXAGONAL KEY USED IN 80 TOOTH

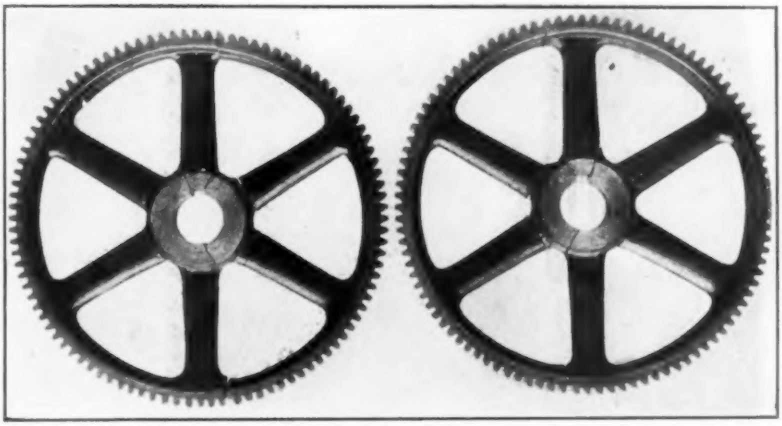


FIG. 22 30 TOOTH ENGAGING 100 TOOTH, TESTS 41 AND 46, SERIES 2

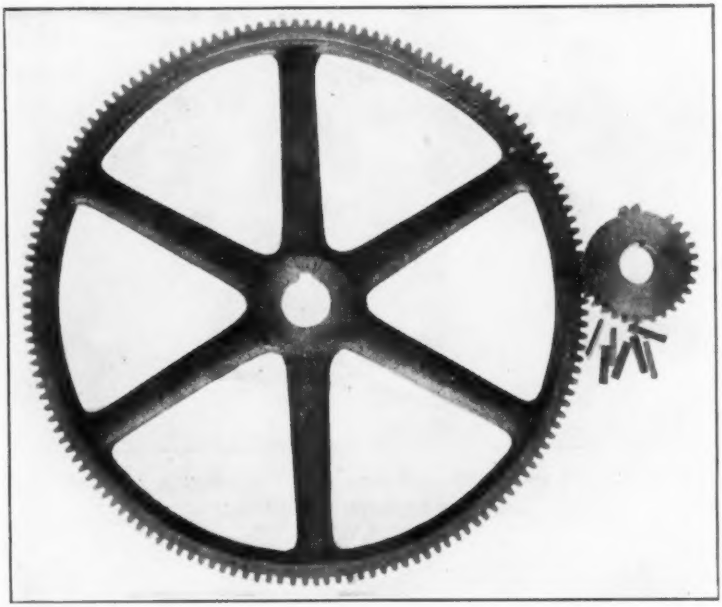


Fig. 23 30 Tooth engaging 150 Tooth, Test 016, Series 1

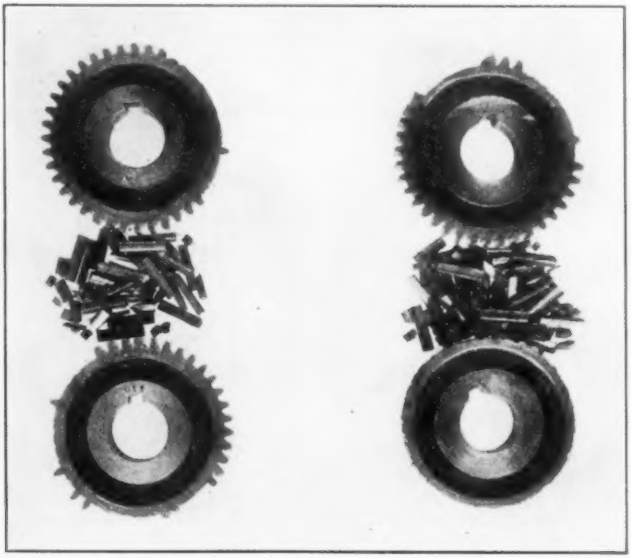


Fig. 24 40 Tooth engaging 40 Tooth, Test 011, Series 1 and 47, Series 2

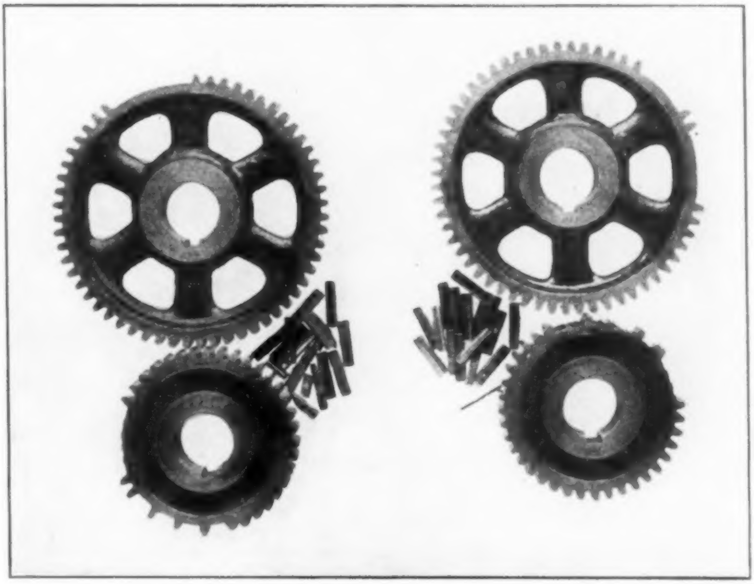


FIG. 25 40 TOOTH ENGAGING 60 TOOTH, TESTS 010 AND 012, SERIES 1

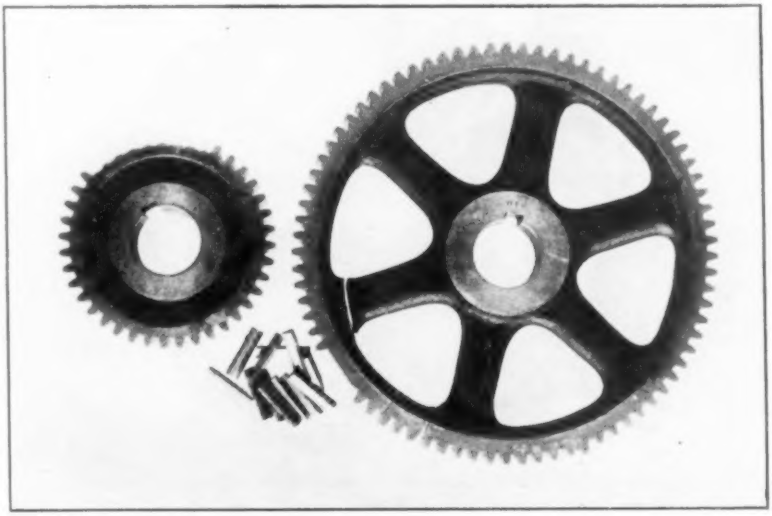


FIG. 26 40 TOOTH ENGAGING 80 TOOTH, TEST 013, SERIES 1

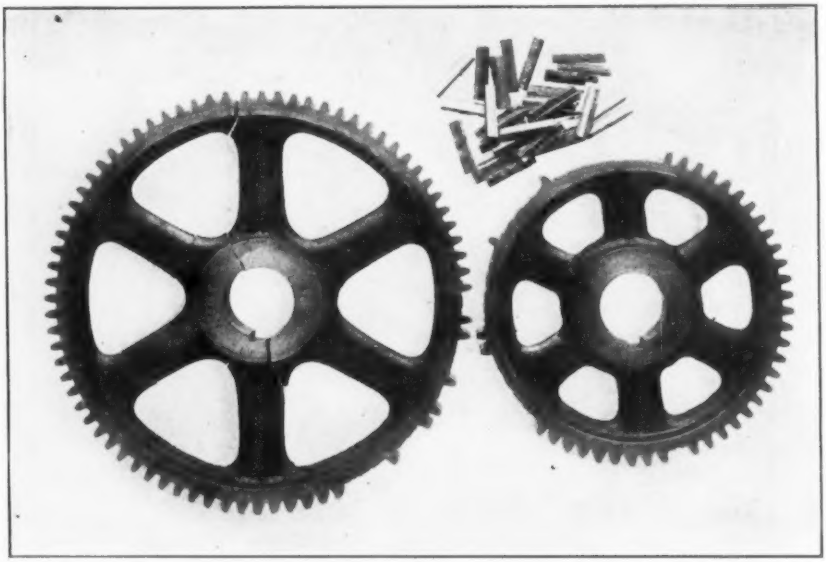


FIG. 27 60 TOOTH ENGAGING 80 TOOTH, TEST 014, SERIES 1

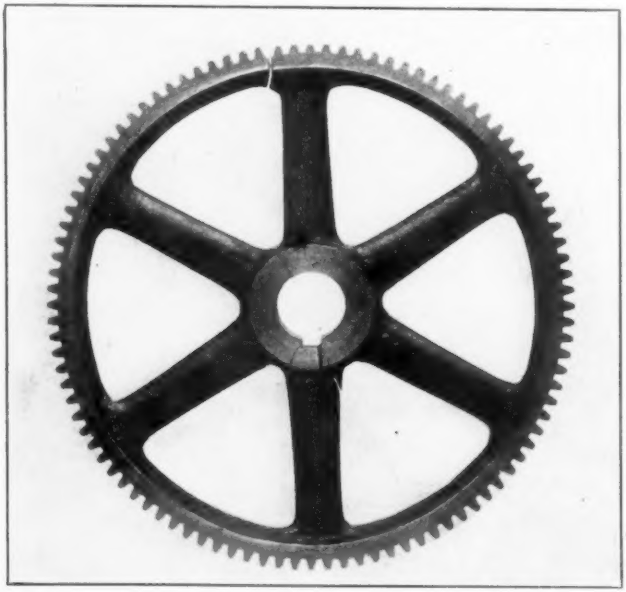


FIG. 28 80 TOOTH ENGAGING 100 TOOTH, TEST 015, SERIES 1. 80 TOOTH INTACT

APPENDIX NO. 1

POINTS FOR FIG. 4, CURVE *B*

By Lewis formula

$$W = spf\left(0.124 - \frac{0.684}{n}\right)$$

$s = 8000$ for 0 to 100 ft. per min.

$W = 8000 \times 0.31416 \times 1.0625 \times 0.90 = 240$............[1]

$s = 4800$ for 300 ft. per min.

$W = 4800 \times 0.31416 \times 1.0625 \times 0.90 = 144$............[2]

$s = 4000$ for 600 ft. per min.

$W = 4000 \times 0.31416 \times 1.0625 \times 0.90 = 120$............[3]

APPENDIX NO. 2

39 To see how closely the factor of strength y obtained by the Lewis formula

$$y = 0.124 - \frac{0.684}{n},$$

corresponded to the forms of teeth used in this system, the following investigations were made:

40 The teeth were measured as accurately as possible, laid out 100 times full size, parabolas inscribed as shown in Figs. 7 and 8, and the distances

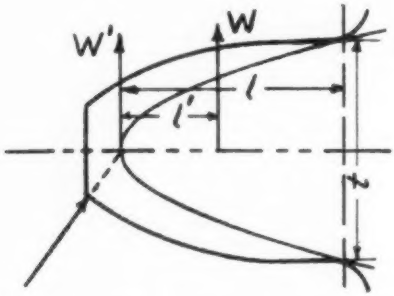


FIG. 29 SHOWING ASSUMED POINT OF APPLICATION OF FORCE AND RESISTANCE OF TOOTH

from vertices to pitch point, and from tangent lines to pitch point, measured. The results are given in Table 16.

41 Referring to Fig. 29, the load is assumed to be applied at the end of the tooth and normal to the profile. The transverse component $= W'$, and the equivalent force at the pitch line $= W$. The thickness of tooth at point of tangency of the inscribed parabola $= t$. The distance from the pitch line to vortex of parabola $= l'$, and from the vortex to the tangent line $= l$.

s = modulus of rupture

f = width of face in inches

$\frac{I}{c}$ = modulus of section

r = pitch radius of gear

Then, from the theory of flexure,

$$W'\,l = \frac{s\,I}{c} = \frac{sft^2}{6} \quad \text{[4]}$$

TABLE 16 DIMENSIONS OF GEAR TEETH

No. of Teeth	Thickness at Depth							End Load Parabola			Pitch Load Parabola		
	0.00	0.048	0.100	0.150	0.175	0.185	0.200	Thicknes at Tangency	Tangent Line to Pitch Line	Vertex to Pitch Line	Thickness at Tangency	Vertex to Pitch Line	Tangent Line to Pitch Line
20	0.070	0.1240	0.1570	0.1630	0.1630	0.1630	0.1680	0.1662	0.0964	0.0760	0.1680	0.0152	0.1002
30	0.080	0.1277	0.1572	0.1678	0.1701	0.1710	0.1760	0.1750	0.0975	0.0740	0.1784	0.0150	0.1014
40	0.086	0.1260	0.1570	0.1730	0.1760	0.1785	0.1830	0.1820	0.0964	0.0797	0.1860	0.0160	0.1013
60	0.092	0.1300	0.1570	0.1745	0.1800	0.1840	0.1910	0.1896	0.0972	0.0800	0.1918	0.0158	0.1002
80	0.086	0.1260	0.1550	0.1760	0.1815	0.1850	0.1910	0.1902	0.0976	0.0789	0.1922	0.0178	0.1005
100	0.088	0.1240	0.1550	0.1750	0.1820	0.1870	0.1950	0.1920	0.0946	0.0815	0.1966	0.0178	0.1015
150	0.096	0.1300	0.1570	0.1800	0.1900	0.1930	0.2000	0.1980	0.0968	0.0815	0.2020	0.0191	0.1020
Rack	0.1053	....	0.1570	....	....	....	0.2094	0.2080	0.0983	0.0864	0.2126	0.202	0.1028

TABLE 17 COMPARISON OF VALUES OF Y

Size of Gear	$\frac{k}{6p}$	Table 1 Lewis	$0.124 - \frac{0.684}{n}$
20T	0.0914	0.090	0.0898
30T	0.0995	0.102	0.1012
40T	0.1038	.....	0.1069
60T	0.1105	0.114	0.1126
80T	0.1109	.....	0.1154
100T	0.1128	0.118	0.1172
150T	0.1180	0.120	0.1194

$$W' = \frac{sft^2}{6\,l} \dots\dots\dots\dots\dots\dots\dots\dots [5]$$

Also, by moments about center of gear,

$$W'\,(r + l') = Wr \dots\dots\dots\dots\dots\dots\dots\dots [6]$$

$$W' = W\,\frac{r}{r+l'} \dots\dots\dots\dots\dots\dots\dots\dots [7]$$

Substituting value of W' from [7] in [5],

$$W\,\frac{r}{r+l'} = \frac{sft^2}{6\,l} \dots\dots\dots\dots\dots\dots\dots\dots [8]$$

$$W = \frac{sf}{6} \cdot \frac{t^2}{l} \cdot \frac{r+l'}{r} \dots\dots\dots\dots\dots\dots\dots\dots [9]$$

Say,

$$W = \frac{sf}{6} \cdot k \dots\dots\dots\dots\dots\dots\dots\dots [10]$$

where

$$k = \frac{t^2}{l} \cdot \frac{r+l'}{r} \dots\dots\dots\dots\dots\dots\dots\dots [11]$$

By Lewis formula,

$$W = spfy$$

Hence,

$$spfy = \frac{sf}{6} \cdot k$$

$$y = \frac{k}{6\,p} \dots\dots\dots\dots\dots\dots\dots\dots [12]$$

42 To find the factors y corresponding to the gear teeth of these tests, computations were made using the actual dimensions as given in Table 16. The results are formulated in Table 17, together with the values of y given in Table 1 of Lewis's original paper, and as computed from his formula

$$y = 0.124 - \frac{0.684}{n}$$

for 15 deg. involute, given in the American Machinist, June 22, 1893.

43 No single expression of the form $x = \frac{y}{n}$ can satisfy all of these values of $\frac{k}{6\,p}$. They call for a range from $0.124 - \frac{0.652}{n}$ to $0.124 - \frac{1.120}{n}$ but the expression $0.124 - \frac{0.684}{n}$ gives values for the factor of safety which do not vary by as much as 5 per cent in any case from the values determined from the actual tooth dimensions.

APPENDIX NO. 3

DETERMINATION OF ARC OF ACTION

44 The tooth comes into contact, neglecting for the present those cases of gears with such a small number of teeth that there would be interference, when the addendum circle of the driven gear cuts the line of action. This position of the tooth is shown at c', Fig. 30.

45 Contact between this tooth and its mate will continue until the addendum circle of the driver cuts the line of action, again neglecting those cases involving interference. This position is shown at c. The length of the arc of action $d'd$ is important as it governs the number of pairs of teeth simultaneously in contact.

46 The position of the tooth for pitch-point contact is shown at a. The corresponding point of tangency of the line of action and the base circle is b. It is obvious that b'' bears this same relationship to d; and b''' to d'. Hence, when the driven gear tooth travels from position $c'd'$ to cd, which is its complete period of action, the base circle will have rotated through the arc $b'''b''$.

From the nature of the involute curve,

$$ab\ (= db'') = cb + \text{arc}\ bb''$$

Also

$$ab = ac + cb$$

Hence

$$ac = \text{arc}\ bb'' \qquad [13]$$

Similarly it can be shown that

$$ac' = \text{arc}\ bb''' \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [14]$$

Adding [13] and [14]

$$ac + ac' = \text{arc}\ b'' + \text{arc}\ b'''$$

Hence $cc' = \text{arc}\ b''\ b''' =$ length of line of actual action.

47 The cases involving interference may also be considered by reference to Fig. 30. If either gear has so few teeth that ab' becomes less than ac', the value of $b''\ b'''$ will, of course, equal $b'c$ (and not $c'c$). If both gears are low-numbered so that ab' is less than ac', and ab less than ac, the value of $b''\ b'''$ will equal $b'b$.

48 It is evident from Fig. 30 that the radius of base circle = the radius of pitch circle × cos 14½ deg. for the 14½ deg. involute system. Hence

Radius of pitch circle = radius of base circle × sec 14½ deg.

And

Arc of pitch circle corresponding to $b''\ b'''$ of base circle = $b''\ b'''$ sec 14½ deg. = length of arc of action.

49 The actual length of the line of action in any case may be readily computed.

50 Referring to Fig. 31, Oac is a triangle in which

side Oa = pitch radius of driver r

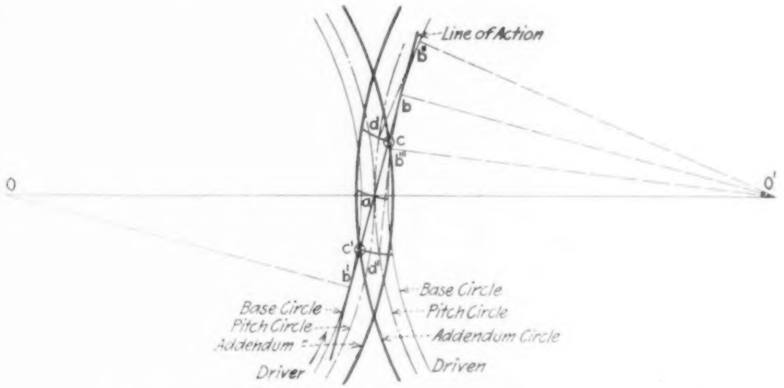


FIG. 30 SHOWING POINT OF INITIAL CONTACT OF TEETH

side Oc = pitch radius of driver plus addendum $r + a$

angle Oac = 90 deg. + 14 deg. 30 min. = 104 deg. 30 min.

$$\sin x = \sin 104 \text{ deg. } 30 \text{ min. } \frac{r}{r+a}$$

hence angle x becomes known.

$$\text{Angle } y = 180 \text{ deg.} - (104 \text{ deg. } 30 \text{ min.} + \text{angle } x)$$

$$ac = r\,\frac{\sin y}{\sin x}$$

Similarly ac' may be computed for the driven gear.

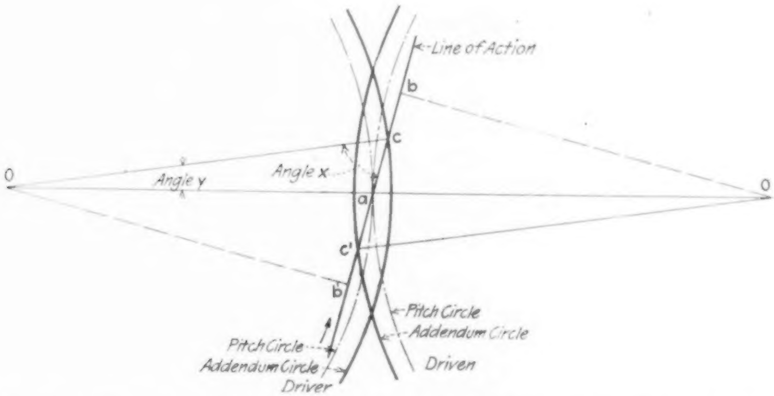


FIG. 31 DIAGRAM TO ILLUSTRATE LENGTH OF LINE OF ACTION OF TEETH

51 In cases involving interference it must be borne in mind that $ab' = 0a \sin 14\frac{1}{2}$ deg. and $ab = 0' a \sin 14\frac{1}{2}$.

TABLE 18-a COMBINATIONS OF 10-PITCH GEARS

Gear, No. Teeth	*ab*, Inches	*ac*, Inches
20	0.25038	0.27182
30	0.37557	0.29605
40	0.50076	0.31208
60	0.75114	0.33246
80	1.00152	0.34495
100	1.25190	0.35345
150	1.87775	0.36637

The 20 tooth is the only one in the set used in these experiments where *ab* is less than *ac*, i. e. where theoretical interference is involved.

TABLE 18-b COMBINATIONS OF 10-PITCH GEARS

Gear Teeth		Line of Contact, Inches	Arc of Contact, Inches
Driver	Driven		
20	30	0.52220	0.53938
20	100	0.52220	0.53938
20	150	0.52220	0.53938
30	30	0.59210	0.61158
30	40	0.60813	0.62814
30	60	0.62851	0.64919
30	80	0.64100	0.66209
30	100	0.64950	0.67087
30	150	0.66332	0.68514
40	40	0.62416	0.64469
40	60	0.64454	0.66575
40	80	0.65703	0.67866
60	80	0.67741	0.69970
80	100	0.69840	0.72138

In all these cases except those involving 20--Tooth gears the line of contact equals the sum of the respective *ac's*. In cases involving the 20-Tooth gears it equals the *ab* of the 20-Tooth plus the *ac* of the 20-Tooth.

52 From the values of *ab*, *ab'*, *ac* and *ac'* so computed the length of the actual line of action in any given case becomes known. This gives the length of the corresponding arc $b'' b'''$ and multiplying this by sec $14\frac{1}{2}$ deg., the actual length of the arc of action is obtained.

53 Tables 18-a and 18-b give the results of such computations for the combinations of 10-pitch gears used in these tests.

APPENDIX NO. 4

DERIVATION OF FORMULA

$$W = spf\left(0.154 - \frac{1.26}{n}\right)$$

54 Assuming the Lewis form of expression

$$W = spf\left(x - \frac{y}{n}\right)$$

as the general formula in which the values of x and y are to be determined.

55 It is known that

$$\begin{aligned} s &= 39000 \text{ in all cases} \\ p &= 0.31416 \text{ in all cases} \\ f &= 1.0625 \text{ in all cases} \\ W &= 1184 \text{ when } n = 20 \\ &= 1451 \text{ when } n = 30 \\ &= 1608 \text{ when } n = 40 \\ spf &= 13018. \end{aligned}$$

When $n = 20$

$$1184 = 13018\left(x - \frac{y}{20}\right) \quad \text{[15]}$$

$$0.091 = \left(x - \frac{y}{20}\right) \quad \text{[15a]}$$

$$1.82 = 20\ x - y \quad \text{[15b]}$$

Similarly when $n = 30$

$$1451 = 13018\left(x - \frac{y}{30}\right) \quad \text{[16]}$$

$$0.112 = x - \frac{y}{30} \quad \text{[16a]}$$

$$3.36 = 30\ x - y \quad \text{[16b]}$$

And, when $n = 40$

$$1608 = 13018\left(x - \frac{y}{30}\right) \quad \text{[17]}$$

$$0.123 = x - \frac{y}{40} \quad \text{[17a]}$$

Subtracting [15b] from [16b]

$$\begin{aligned} 30\ x - y &= 3.36 \\ 20\ x - y &= 1.82 \\ \hline 10\ x &= 1.54 \\ x &= 0.154 \end{aligned}$$

Substituting in [16*a*]

$$0.091 = 0.154 - \frac{y}{20}$$

$$\frac{y}{20} = 0.063$$

$$y = 1.26$$

Therefore

$$x - \frac{y}{n} = 0.154 - \frac{1.26}{n}$$

Check

$$0.154 - \frac{1.26}{20} = 0.091 \quad \text{when } n = 20$$

$$0.154 - \frac{1.26}{30} = 0.112 \quad \text{when } n = 30$$

$$0.154 - \frac{1.26}{40} = 0.1225 \text{ when } n = 40$$

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

CONTENTS

FOREIGN REVIEW

With this number the Foreign Review enters into its second year. The considerable number of inquiries received in connection with this new branch of the activities of the Society and the fact that several large firms having research departments have made special arrangements for handling the material published in the Foreign Review, shows that it is rendering a needed service.

THIS MONTH'S ARTICLES

Among this month's articles will be found two interesting communications on air machinery: tests of a turbo-compressor and of tunnel compressed air locomotives; the latter of particular interest owing to the scarcity of experimental data on the efficiency of this kind of apparatus due to the great inconvenience of testing under working conditions. Ventou-Duclaux's article on the use of naphthalene for internal combustion engines describes several types of French carbureters for use in this class of machinery, and gives data of tests. The naphthalene engine is of certain interest as being one of the cheapest in operation, and nearly the safest; its main disadvantage lies in the fact of its being very far from fool-proof. Heinz's article on peat utilization indicates that peat gas power is already the cheapest source of power in existence. Several new types of internal combustion engines are described, particular attention being here called to the Urbani two-stroke cycle engine.

In the Machine Shop section is given a description of a new crucible furnace, economical in fuel consumption and life of crucibles, as well as interesting data on the cost of heat treatment of metals by gas; an article by Buderus on the testing of molding sands is also of interest.

The article on the Kommer gear in the November issue of The Journal brought forth several inquiries for more information, in reply to which an abstract of the second part of the Kommer

article is here given describing another type of the Kommer gear. Attention is also called to the article by Wolzogen Kühr on the design of centrifugal fans without using Euler's formulae.

In the section of Steam Engineering will be found several articles on the design of high capacity boilers, including the Glogner type, as well as La Maestra's article on the corrosion in boilers giving formulae for calculating the safe life of a boiler after some corrosion has started, as well as a table of what the author calls "time coefficients for corrosion in boilers." New processes for determining the heat conductivity in plate shaped bodies are given, of higher reliability than the methods previously used.

The section Miscellanea covers such articles as are of interest to the American mechanical engineer, but cannot through lack of space, be given a regular section in the Foreign Review. Among this month's abstracts will be found several articles in connection with farm machinery, automatic fire protective apparatus, as well as a number of smaller abstracts.

The Editor will be pleased to receive inquiries for further information in connection with articles reported in the review. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Aeronautics

AUTOMATIC RESTORATION OF EQUILIBRIUM OF FLYING MACHINES (*Zur selbsttätigen Bekampfung von Gleichgewichtsstörungen bei Flugzeugen*, z.D.K.N. *Prometheus*, vol. 24, no. 1204, p. 116, November 23, 1912, 2 pp. *t*). The author, evidently connected in some way with the German military aviation corps, recommends a system of *automatic stabilization* based on the following principles: (*a*) separation of the airship into two rigidly connected parts, one comprising the carrying and guiding planes, "planes-body," and the other the usual heavy parts, "weight-body"; (*b*) spring joint connection between the two, such that when an external action produces change in the mutual position of the two component parts, this change itself is used to set into action the apparatus used for restoring the original state; (*c*) arrangements for cutting out the automatic stabilizing arrangement when required, e.g. in landing; (*d*) independence of the individual steering devices from those providing automatic stabilization. The author attempts to prove that the separation of the airship into two parts, "weight-body" and "planes-body" would contribute to an easier

installation of automatic stabilizing devices. The arrangement proposed has never been tried.

A SPEED FORMULA APPLICABLE TO AEROPLANES (*Sur une formule de vitesse applicable aux aéroplanes*, A. Berget. *Comptes rendus des séances de l'Académie des Sciences*, vol. 155, no. 20, p. 963, November 11, 1912. 2 pp. *eA*). The author derived the following experimental formula for the determination of the speed of an aeroplane:

$$V = A\sqrt[3]{\frac{F}{S}}$$

where V is the speed of the aeroplane in myriameters (1 myriameter= 6.2137 miles) per hour, S the supporting surface in sq. m., F power of the motor in h.p. (continental h.p. equals 736/746 of American h.p.), while A is a numerical coefficient whose value lies between 7 and 8. The value of A has been established experimentally, as well as the formula itself, from an investigation of a number of aeroplanes of various systems, and takes care of the elements of aeroplane design not otherwise considered in this formula. The best results are obtained when the value of A is 8 or very close to 8, but it must not be accepted from this that if an aeroplane has A equal to 7 or but slightly in excess of 7, it is unsatisfactory: that may simply mean that it has not been designed for speed, or that speed has been considered of secondary importance as compared with safety and strength of construction. A table of data for various French aeroplane types is given in the article showing that in all cases A lies practically between 7 and 8.

Air Machinery

TESTS OF A TURBO-COMPRESSOR AT THE WESTERHOLT MINE. (*Untersuchung eines Turborcompressors auf der Zeche Westerholt. Glückauf*, vol. 48, no. 47, p. 1913, November 23, 1912. 3 pp., 2 figs. *de*). Description and data of *tests of an air compressor plant* at the Westerholt mine, Ruhr district, Prussia, from a communication of the Essen office of the Association for the Inspection of Steam Boilers. The turbine driving the compressor is so arranged as to be able to work both with live and exhaust steam, while the compressor can take in either 10000 cbm (say 35000 cu. ft.) of air per hour and compress it to 6 atmospheres, itself running at 4400 r.p.m., or it can run at 4200 r.p.m., and compress to 5.5 atmospheres 5000 cbm of outside air. The consumption of steam was guaranteed to be: for exhaust steam only, when handling 10000 cbm, 1.4 kg/cbm (say 0.04 lb. per cu. ft.), for 5000 cbm, 1.78 kg/cbm (say 0.051 lb. per cu. ft.); for live steam only 0.69 kg/cbm and 0.88 (say 0.02 and 0.025 lb. per cu. ft.) for the full and half load respectively. The actual test values are slightly more favorable than these. The article contains full data of the test.

TESTS OF TUNNEL COMPRESSED AIR LOCOMOTIVES (*Untersuchungen an Tunnel-Druckluftlokomotiven*, V. Litz. *Glückauf*, vol. 48, no. 45, p. 1825, November 9, 1912. 7 pp., 4 figs. *eA*). For the construction of the Mont d'Or tunnel the A. Borsig works, of Berlin-Tegel, Germany, have delivered two types of compressed air locomotives, three-axle of 11 tons in working order, and four-axle of 31 tons weight, both with double preheating of

air done in the following manner: The air after coming from the pressure reducing valve between the compressed air tanks on the locomotive and the high-pressure cylinder passes through a special set of heater pipes, and again through a second set of similar pipes in passing from the high-pressure to the low-pressure cylinder. The furnace for heating the pipes is on the locomotive frame, a small stack being provided in the front part of the locomotive to take care of the gases of combustion. Coal and charcoal are used for fuel in order to obtain a practically smokeless combustion. The preheating is such as to raise the temperature of the air in both cylinders to about 180 deg. cent. (356 deg. fahr.). With a grade of 0.13 per cent, the larger locomotives can haul up to 180 tons, and the smaller up to 55 tons gross (in all cases a metric ton of 2200 lb. is meant). Fig. 1 shows the general arrangement of the locomotive.

The data are interesting particularly on account of the scarcity of tests of tunnel locomotives under operative conditions, tunnel construction being

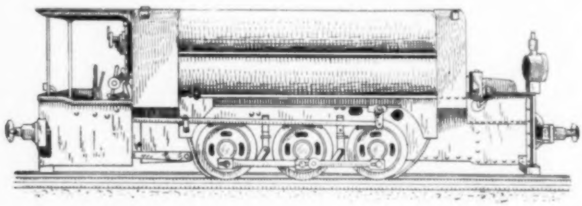


FIG. 1 BORSIG COMPRESSED AIR LOCOMOTIVE

usually done under conditions and guarantees as to the time of delivery which permit no interruptions for the purposes of tests. During tests the locomotive was in front of the train when it moved tunnelwards, and behind when it moved out of the tunnel; in normal operation the opposite arrangement is of course used, since with a grade of 0.13 per cent and the heavy trains used, a break of a coupling would cause great danger unless the locomotive was always below grade from the train. To take care of that during the tests, there was always a second locomotive at some distance on the other side of the train. Both locomotives have been tested just as they were, without previous overhauling or even cleaning. Previous to the tests they were over a year in actual service on the construction. The air capacity of the tanks could be established only by calculation, and was found to be for the small locomotive 2250 l. (78.75 cu. ft.), and for the large locomotive 10200 l. (357 cu. ft.). The comparatively low temperature of preheating indicates that the smoke stack was chocked up, the temperatures obtained at the delivery tests having been considerably higher. The author states, however, that temperatures of 45 to 54 deg. cent. (113 to 129.2 deg. fahr.) in front of the low-pressure cylinder are about as high as one can go economically with this kind of locomotive where strong preheating permits the expansion of the air to be driven very high in the cylinder. The data of the tests are given in Tables 1 and 2.

See also Pumps and Fans.

TABLE 1 COMPRESSED AIR TUNNEL LOCOMOTIVES TEST

	Length of Run, m.	Gross Load, t.	Output, tkm.	Useful Load, t.	Tractive Effort Locomotive and Train, Kg.	Output I. H. P., Hr.	Consumption of Air: Total Cbm.	Consumption of Air: Per Tkm, Cbm.	Consumption of Air: Per H.p-Hr., Cbm.
	European Units								
Small Locomotive:									
Upgrade	1000	76.35	76.35	55.80	1627	6026	123.75	1.621	24.317
Downgrade	1000	76.35	76.35	55.80	60	0/222	4.50	0.059	24.324
	American Units								
	Miles	Tons Am.	Ton-Miles	Tons Am.	Lb.		Cu. Ft.	Per Ton-Mile, Cu. Ft.	Per H.p. Am.-Hr., Cu. Ft.
Upgrade	0.621	69.4	43.09	50.7	3579		4332	32	840
Downgrade	0.621	69.4	43.09	50.7	132		158	1.22	840
	European Units								
Large Locomotive:									
Upgrade	1870	183.20	324.58	138.2	3645	25245	459.00	1.34	22.611
Downgrade	1770	183.20	324.26	138.2	100	0.656	13.26	0.04	23.806
	American Units								
Upgrade	1.16	166.3	192.90	125.8	8019		16065	26.3	783
Downgrade	1.09	166.3	181.267	114.2	220		455	0.79	825

Cranes

New Crane Constructions for Special Purposes (*Neue Kranbauarten für Sonderzwecke*, C. Michenfelder. *Zeits. des Vereines deutscher Ingeniure*, vol. 56, no. 41, p. 1645, October 12, 1912. 5 pp., 15 figs. *d*). Description of some new types of cranes, mainly those for special purposes: thus there is a description of a *pickling* crane for lowering metal articles into a pickling bath and taking them out of it, where the difficulty lies in preventing the crane installation from being affected by the vapors of the pickling bath; it was found in this respect that with properly designed and enclosed motors, and sufficient ventilating and exhaust arrangement in the pickling room, there is little trouble due to the action of the

TABLE 2 GENERAL DATA OF LOCOMOTIVES

	Small	Large
Number of cars	9	15
Kind of cars	box cars	15 tip wagons, 1 box car
Light weight	2.2 tons 4840 lb.	2.8 or 2.2 ton 6160 or 4840 lb.
Gross weight	8.4 tons 18480 lb.	11.6 or 8.4 ton 25520 or 18480
Duration of run, min	10.5	20.5
Length of run, m/yards	1000/1093	1870/2043
Speed m. per sec./ft. per sec	1.59/4.77	1.52/4.65
Initial pressure in the air tanks, atmospheres	84	68
Final pressure in the air tanks, atmospheres	29	23
Number of readings taken	10	19
Grand average pressure in the working flask, atmospheres	15	15.5
Temperature of the entering air at beginning of run:		
high-pressure cylinder, cent./fahr	....	80/176
low-pressure cylinder, cent./fahr	....	80/176
Temperature of entering air at end of run:		
high-pressure cylinder, cent./fahr	80/176	64/147.2
low-pressure cylinder, cent./fahr	45/113	58/136.4

The above are data for the run upgrade; the original article contains similar data for the downgrade run as well.

gases. The article describes also some of the arrangements of the pickling room. The other kind of crane described in considerable detail is for ice-making plants.

Internal Combustion Engines

Tests of an Internal Combustion Engine (*Versuche an einem Verbrennungsmotor*, Fritz L. Richter. *Dinglers polytechnisches Journal*, vol. 327, nos. 44 and 45, pp. 693 and 710, November 2 and 7, 1912. 7 pp., 18 figs. *et*). Description of and data from *tests of an internal-combustion engine* made at the machine laboratory of the Royal Technical School at Chemnitz. The tests have shown that, when an *ordinary indicator* is used, the *friction losses* are so large as to make the data entirely unreliable; both during compression and expansion the pencil is behind the

position where it ought to be, and the diagram is too large on both sides. There are also considerable difficulties in correctly determining the *temperatures*, especially that of the exhaust gases. The article contains many determinations of the tractive power of the engine, showing among other things how it and the efficiency of the engine are influenced by the cylinder lubrication.

ARIS CARBURETER (*Le carburateur Aris. La Pratique automobile*, vol. 8, no. 178, p. 3795, November 25, 1912. 2 pp., 5 figs. *d*). Description with several illustrations of the *Aris carbureter*. It has no float chamber. The fuel is admitted first to a horizontal passage provided with a needle valve, and then to a vertical passage at right angles to the first. This passage is also closed by a needle connected with a rotatable valve closing the passage from the air chamber of the carbureter to the cylinder in such a manner that it opens the passage to the fuel in proportion as the suction in the cylinder forces the valve up. The amount of fuel admitted may therefore be kept in constant ratio to that of air no matter what the speed of the motor may be, at least within certain limits, and the mixture remains practically always of the same richness.

NAPHTHALENE ENGINES (*Les moteurs à naphtaline*, L. Ventou-Duclaux. *La Technique moderne*, vol. 5, no. 9, p. 305, November 1, 1912. 2 pp., 2 figs. *d*).

UTILIZATION OF NAPHTHALENE AS A FUEL IN EXPLOSION MOTORS (*Utilization de la naphtaline comme combustible dans les moteurs à explosion. La Revue électrique*, vol. 18, no. 213, p. 376, November 8, 1912. 2 pp. *pt*). From a paper read on October 18, 1912, before the French Society of Civil Engineers by L. Ventou-Duclaux on the use of *naphthalene* in internal-combustion *engines* and the design of engines for this fuel. Naphthalene is one of the products of tar distillation separating in the region of 170 to 230 deg. cent. (338 to 446 deg. fahr.) although the usual boiling point is stated to be 217 deg. cent. (422.6 deg. fahr.). It is a solid having a melting point of 79 deg. cent. (174.2 deg. fahr.). To produce an explosive mixture in the cylinder of the engine, one must therefore not only liquefy it, but impart to it a fluidity comparable to that of other hydrocarbons. This is done either by utilizing the heat of cooling water, or by that of the exhaust gases; the piping carrying melted naphthalene to the carbureter being so arranged that its temperature should not go below 80 deg. cent. (176 deg. fahr.), in order to prevent possible crystallization of the naphthalene.

There are two methods of using naphthalene: dissolved in another hydrocarbon (benzole, gasolene, alcohol, etc.), or molten. On the first process is based the Rutgers carbureter, having an arrangement for producing the solution gradually, as it is wanted, so as not to cause any unnecessary deposits of naphthalene; so far, this process has not proved practical. The other process is represented by several designs of carbureters, some using the heat of the cooling water for melting the naphthalene (Deutz and the Schneider Company), while others (Chénier and Lion) use the heat of the exhaust gases. The latter two make the carbureter proper and the melting pot in one piece, and thus prevent the cooling of the molten mass in pass-

ing from the pot to the carbureter. In the Burlat carbureter the melting pot is separate from the carbureter proper, but the connecting pipe is surrounded by exhaust gases, so that the naphthalene reaches the carbureter at a comparatively high temperature. In the Bruneau carbureter both the melting pot and the carbureter proper are surrounded by the same jacket traversed by hot exhaust gases; the air for the explosive mixture is also preheated. In the Lion carbureter a sort of fractional distillation is used: naphthalene which does not have to be particularly pure in this case is first heated to the melting point in one reservoir, then passed on to a second reservoir where it is heated up to the boiling point, and admitted to the carbureter proper in the form of a gas; all impurities, such as tar, anthracen, tar oil, etc., having a higher range of distillation, simply remain in one of the reservoirs, and are cleaned away from time to time.

A 5-hr. test of an 8-h. p. engine, running at 595 r.p.m., showed a consumption of 0.342 kg. (0.75 lb.) per h.p.-hr., an interesting figure considering the low present price of naphthalene.

Besides its low price, the advantages of naphthalene are: safety of operation due to the fact that naphthalene is practically uninflammable and that should there appear a leak, some naphthalene would flow out, immediately solidify in the colder outside air, and stop all further leaks, thus preventing the danger so familiar to all users of gasolene; its chemical constituency is practically constant, thus considerably facilitating the engine design, and raising the average efficiency of its operation (it is a well-known fact that an engine designed for a certain kind of fuel and working with one which may have the same trade name, but different technical properties, may give quite unsatisfactory results). The disadvantage of the naphthalene motor lies in the complications at starting, but even this has little importance in the case of engines having a long average period of operation. The Schneider Company is proposing to use it for locomotive engines on the Transsiberian Railroad where steam locomotives have proved unsatisfactory owing to the large amount of mineral salts (up to 5 per cent) in the available feedwater.

MIXED EXPLOSION OR COMBUSTION AND COMPRESSED AIR ENGINES (*Moteurs mixtes à explosion ou à combustion et à air comprimé*, L. Letombe. *Mémoires de la Société des Ingénieurs Civils de France*, 7th ser., vol. 65, no. 359. 24 pp., 14 figs. *d.* Same article reprinted nearly in full in *La Revue électrique*, vol. 18, no. 214, p. 418, November 22, 1912). The title is somewhat misleading; what the author mainly discusses is the use of compressed air for self-starting. He describes his supplementary valve for the *self starter valve gear*, as well as an *air compressor* with an automatic arrangement for throwing it in and out of engagement.

UTILIZATION OF PEAT BOGS FOR THE PRODUCTION OF POWER AND ITS INFLUENCE ON THE DEVELOPMENT OF AGRICULTURE (*Die Ausnutzung der Torfmoore zur Krafterzeugung und ihr Einfluss auf die Kulturentwicklung*, C. Heinz. *Die Gasmotorentechnik*, vol. 12, no. 8, p. 121, November, 1912. 8 pp., 5 figs. *dg*). The article describes the Görlitz peat gas producer (cp. *The Journal*, July 1912, p. 1101) and special producer gas engine designed

to use peat producer gas, as well as discusses the general problem of peat utilization in its relation to power production and agriculture. The author quotes the following data as to the cost of peat per ton in Germany (of 2200 lb.) in marks (1 mark = $0.238): wages 0.26, power 0.24, drying 0.70, maintenance of plant 0.20, supervision 0.20, delivery cost 0.40, depreciation 0.50, total per ton of air dried peat: marks 2.50, and since 1 kg is required for the production of one effective horsepower-hour, the fuel cost of the latter is ¼ pf., or roughly 1/16 cent. Since, further, depreciation, interest, and other costs of a peat gas plant do not differ materially from those of other heat engine power plant costs, the peat gas power plant appears to be the cheapest in existence. Tests made by the Silesian Steam Engine Inspection Association show that as much power is derived from 1 kg peat in a peat gas producer engine as from 1 kg coal in a steam engine, while peat is available in many localities where coal is scarce and is much cheaper. The author also calls attention to the fact that the utilization of peat bogs for power production has the further advantage eventually of converting a useless, often harmful, bog into valuable agricultural land.

Motors at the Exhibition (*Les moteurs à l'Exposition*, P. James. *L'Aérophile*, vol. 20, no. 22, p. 512, November 15, 1912. 5 pp., 28 figs. *d*). Description of aeronautic engines at the Paris Salon of 1912. In the *Dhenain engine* (Fig. 2A) the valve gear is controlled by an eccentric in combination with a pinion of peculiar construction. By turning the eccentric this pinion is displaced, and so the admission, exhaust and direction of rotation of the engine may be regulated while the engine is running. The cranks are interchangeable, work all in the same plane, with the crankheads helicoidally arranged. The *Sira engine* (Fig. 2B) has eight cylinders arranged starwise, but neither cranks nor connecting rods. The piston gear is operated by two special cams, with a single rod. The *Esselbé engine* is original in design. The cylinders are replaced by a tore in which move diametrically opposite pistons A A', B B', each pair fixed on a plate CD. (Fig. 2C and D). These pistons open and close in a manner similar to that of shears. The space between the tore and the four pistons constitutes four chambers of which volumetric variations correspond to the four phases of a four-stroke cycle, the tore itself also rotating and thereby bringing the ports p and q of admission and exhaust to the places where they belong at the right times. To transform the reciprocating motion of the plates C and D into rotary, they are made solid with two pipes S and T concentrically around the axis of the engine and carrying at their other extremity two beams in articulated connection with the cranks E; these cranks actuate the central pinion of the tore. The article gives no data as to the efficiency of this engine. For the Esselbé engine Cp., also The Automobile, December 19, 1912, p. 1264.

A New Two-Stroke Cycle Engine (*Un nuovo motore a due tempi*. Marino Urbani. *L'Industria*, vol. 26, no. 44, p. 695, November 3, 1912. 1 p., 3 figs. *d*). Description of the *Urbani two-stroke cycle engine*. It consists essentially of the two cylinders 1 and 1′ (Fig. 3) joined in-U; one of them is provided with the exhaust port 2, the other with the openings for

scavenging air 3 and gas 4. The air and gas pumps, which may also be independent of one another, are in the carters 5 and 5′. In the two cylinders move two pistons in full accord with each other having equal cranks making the same angles with the crankshaft. The action of the engine is simple. Suppose that the pistons are at the right-hand side dead points, and that to the right of them is a suitably compressed explosive mixture. When this is ignited an expansion follows, and the pistons move to the left. The exhaust ports 2 open first, and the pressure in the cylinder

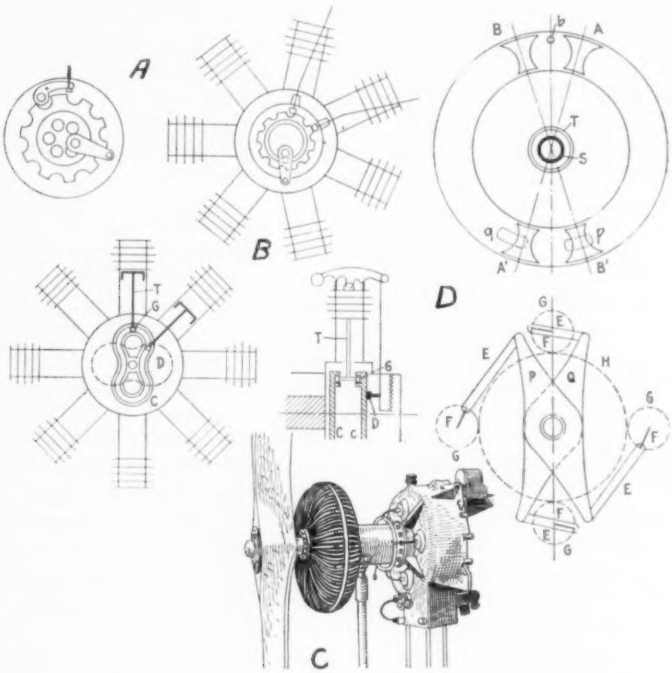


Fig. 2 *A* Drénain Engine; *B* Siva Engine; *C* and *D* Esselbe Engine

rapidly falls to about 0.3 to 0.4 kg/qcm (4.2 to 5.6 lb. per sq. in.) above atmospheric. At this point of the path of the piston the port 3 opens, admitting scavenging air previously compressed in 5′ to a pressure slightly above 0.3 to 0.4 kg/qcm (so as not to produce eddies in the cylinders). Still later, when the pressure in the cylinder has had time to fall below 0.3 to 0.4 kg/qcm, the gas comes through duct 7 and port 4 from pump 5 compressed to somewhat more than the pressure prevailing at that moment in the cylinder, so as to produce eddies and thus help the gas to make a thorough mixture with the scavenging air. On the back stroke of the piston, the ports close in the reverse order, first the gas, then the air, and finally the exhaust. The process is somewhat similar to that of the Oechelhäuser motor, but the engine may be made double acting, tandem, etc.

The air regulator consists of a little piston 8 moving in the cylinder 9 having no head but provided in the middle with a circular passage 10 communicating with 5′. This little piston has an eccentric lagging 90 deg. with respect to the main crank. While the crank makes the angle 0 to 180 deg. corresponding to the period of intake of air, the piston oscillating between the positions *a* and *b*, lets open the annular passage 10 by which air penetrates to 5′. During the compression of the air, while the main crank passes through the angle 180 to 360 deg., the eccentric describes an angle 90 to 270 deg., and the little piston oscillates between *a* and *c*

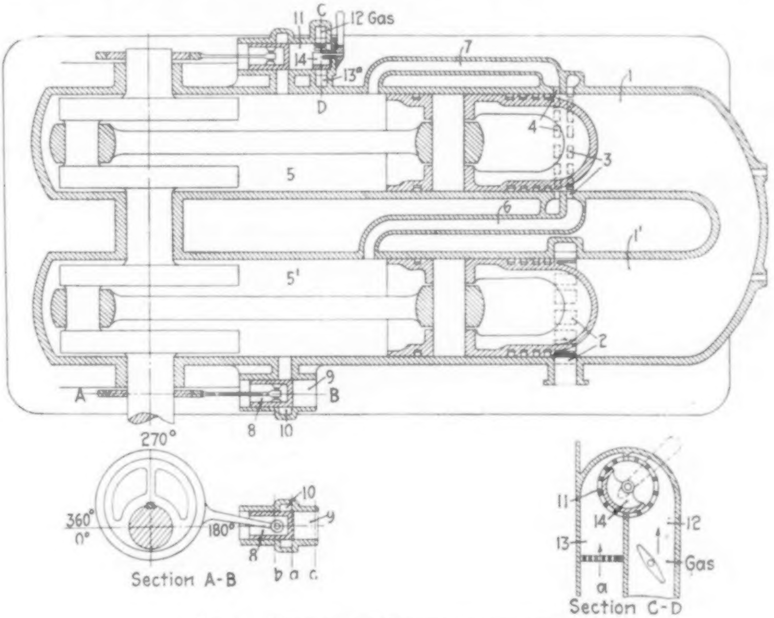


FIG. 3 URBANI TWO-STROKE CYCLE ENGINE

regulating the communication between 5′ and the atmosphere. This arrangement is interesting in that the opening and closing of the annular passage 10 occur always when the piston 8 is at its maximum velocity. A somewhat similar arrangement is provided with respect to gas.

THE ERGON-KOSMOS PRODUCER GAS LOCOMOBILE AND ITS MANAGEMENT (*Die Ergon-Kosmos-Generatorgaslokomobile und deren Behandlung*, W. *Der praktische Maschinen-Konstrukteur*, vol. 45, no. 23, p. 392, November 7, 1912. 2 pp., 1 fig. *d*). Description of the Ergon-Kosmos locomobile, a stationary plant combining in one unit the gas producer, cleaner and locomobile.

Machine Shop

TESTING AND ESTIMATION OF MOLDING SAND (*Prüfung und Bewertung des Formsandes*, Carl Buderus. *Giesserei-Zeitung*, vol. 9, no. 20, p. 629, October 15, and p. 658, November 1, 1912. 6 pp., 1 fig. *pt*). After discussing

the various stresses to which sand is subjected in the mold after the pouring of the metal, the author comes to the conclusion that its power of resisting them depends on what he terms its *plasticity*, and the whole problem of testing molding sand thus becomes one of finding a method for measuring plasticity.

If a sand cylinder of length of axis h and diameter D has the ability not to break when unsupported, and if $\frac{h}{D} = \frac{1}{4}$, then this may be taken as a unit of plasticity which increases with D and falls off with h. In testing plasticity, it is well to keep D constant, and to take h equal to $0.25D$; the plasticity is then determined from the pressure above atmospheric $\frac{pgr}{qcm}$ (p in grams per sq. cm) at which the sand column breaks, this being done in the following manner: if a sand cylinder of $h = 0.25D$ can maintain its shape unsupported at atmospheric pressure, but not at any above it, and if its plasticity be denoted by bi, its specific weight (in the column, i. e. compressed) by s, then

$$bi = 1 = bi_1 = \frac{r^2\pi hs}{\frac{r^2\pi h^2}{6}} = \frac{6s}{h}$$

since $\frac{r^2\pi h^2}{6}$ is the moment of resistance of the cylindrical column. If now a sand column has to support not only its own weight, but also an additional pressure $\frac{pgr}{qcm}$ its plasticity must be equal to

$$bi_2 = \frac{(r^2\pi \,.\, h's + r^2\pi p)}{r^2\pi \,.\, h'^2/6} = \frac{6\,(h's+p)}{h'^2}$$

which, to be expressed in the same units, must be divided by the value of bi_1, and this gives

$$bi_3 = \frac{6(h's+p)}{h'^2} : \frac{6s}{h} = \frac{(h's+p)h}{h'^2 s} = \frac{h^2 s + ph}{h^2 s} = 1 + \frac{p}{hs}$$

the last two values being for the case of $h' = h$. Since, however, the specific weights of various sands are very different from one another, either of the above expressions may give incorrect values. Thus, the plasticity of light sand, if expressed by one of the two latter equations would give too high values. The author recommends therefore the expression

$$bi = bi_2 = bi_1 \times bi_3$$

If this expression is used the hardness and toughness of the sand may be estimated from the power applied in stamping the sand, and its compression determined by separate experiments. The fire resistance of the sand, its permeability to gases, formation of the gases in the sand itself and its tendency to be fritted have also to be determined by separate experiments. In the second part of the article the author describes an apparatus for testing molding sand in which a layer of such sand, compressed to a certain amount may be subjected either to the action of gravity alone, or to an

additional gas pressure as well. The apparatus appears to be simple in operation, cheap and easily handled.

A New Crucible Furnace (*Ein neuer Tiegelofen*, W. H. *Eisen-Zeitung*, vol. 33, no. 46, p. 809, November 16, 1912. 1 p., 1 fig. *d*). Description of a new crucible furnace invented by F. Lange in Memmingen, Germany. The furnace (Fig. 4) is contained in a sheet iron jacket protected by a fire-

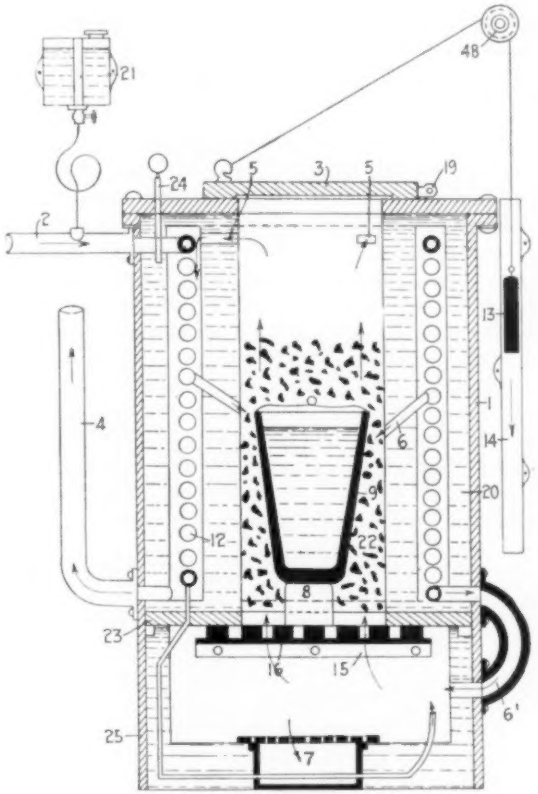


Fig. 4 Lange Crucible Furnace

clay lining. This lining encloses a cylindrical chamber 11 containing a closely wound spiral pipe 12. In the center of the lined shaft is the cylindrical shaft for the reception of the crucible 9 and the coke charge 22 surrounding it. There are four small nozzles 6 opening in the direction of the crucible. Before passing out through stack 4, exhaust gases pass through the ducts 5 into the cylindrical chamber 11 where they preheat the pipes 12. From the top the crucible shaft is closed by the cover 3 rotatable about the hinge 19, the opening and closing of the shaft cover being further facilitated by the rope and counterweight arrangement, as

shown at 13. The whole installation is placed on a bed plate 23 with a round hole in the middle for the admission of air to the crucible shaft; under the plate are located the grates 16 and the angle irons 15 supporting them. A small diameter pipe 25 goes from 12 through the water reservoir 7; the heat of this pipe brings the water up to the boiling point, and the steam, after having mixed with air saturated with kerosene vapors and coming from the pipe 6', passes into the crucible shaft and produces there an intense heat (the air becomes saturated with kerosene vapors while passing through the vessel 21). In spite of the intense heat produced, the consumption of coke is said to be small, while the life of the crucible which in this furnace does not come in contact with cold air at all, is materially longer than in other furnaces.

Annealing and Hardening by Gas (*Glühen und Härten mit Gas. Journal für Gasbeleuchtung*, vol. 55, no. 43, p. 1052, October 26, 1912. 2 pp., 4 figs. *ceA*). Communication from the Stockholm municipal gas plant showing that gas is more economical and convenient than coal for such shop operations as annealing, hardening, tempering, etc. The tests were made in 1911 by engineer H. O. Ödlund at the works of the Separator Company, under regular shop conditions by men usually employed at the plant. Each test lasted a whole day, so as to keep the conditions as uniform as possible. The gas used was supplied by the municipal gas works, and had the lower heat value of 4700 WE (17800 b.t.u. per lb.), and a pressure of 40 mm (1.57 in.) water; for purposes of calculation the price of 11.2 Pf. per cbm ($0.77 per 1000 cu. ft.) has been accepted, this being the present actual rate there. The gas coke had a lower heating value of 7000 WE (27720 b.t.u. per lb.), and cost M 1.40 per 1 hl = 0.1 cbm = 3.5 cu. ft.). The coal was Silkworths blacksmith coal with lower heating value 7000 WE (27720 b.t.u. per lb.), and cost M 1.91 per hl.

Table 3 gives the comparative results with all three fuels. The article also contains descriptions of some of the furnaces used.

Mechanics

Reduction Gear for Coaxial Shafts (*Reduziergetrieb für gleichachsige Wellen*, R. Kommer. *Der Motorwagen*, vol. 15, no. 26, p. 618, September 20, 1912. 3 pp., 5 figs. *d*). The first part of this article has been abstracted in *The Journal*, November 1912, p. 1890. The second part containing a description of another type of the Kommer planetary gear, is here given as a reply to inquiries received. The gear shown in Fig. 5 consists essentially of the stationary external spurwheel a with which is in engagement an internal spurwheel b rotating about a crank pin or eccentric k placed centrically to the stationary spurwheel. The diameter of the wheel b exceeds that of the stationary wheel by twice the crank radius of the shaft k. With the inner spurwheel b is rigidly connected a second inner spurwheel c with which is in engagement the spurwheel d rigidly connected with the driven shaft. The spurwheel d is half as long as the crank radius of the shaft k. In the following description the radii of the wheels will be denoted by r, the numbers of teeth by z, with corresponding subscripts.

By analysing the mutual action of each pair of spurwheels, the author

shows that in the action of the entire gear there correspond to the rotation of the crank k through an angle a: (a) a parallel sliding motion of the spur wheel c; (b) rotation of the toothed wheel d through the angle

$$\frac{a(z_b - z_a)}{z_b}$$

To the parallel sliding motion of the spurwheel c corresponds a rotation of the wheel d through an angle

$$\frac{a(z_d - z_c)}{z_d}$$

To the rotation of the wheel c corresponds a rotation of the wheel d through an angle

TABLE 3 COST OF HARDENING WITH DIFFERENT FUELS

Article	Fuel	Actual Duration of Work per Piece, Sec.	Total Duration of Work per Piece, Sec.	Cost of Fuel per Piece, Pf.	Cost of Labor per Piece, Pf.[1]	Total Cost per Piece, Pf.
Mandrel	Gas	7.6	8.0	0.048	0.125	0.173
	Coke	13.7	16.3	0.098	0.254	0.352
	Coal	12.5	18.1	0.127	0.282	0.409
BearingPins	Gas	5.0	6.3	0.019	0.099	0.118
		8.1	9.6	0.054	0.149	0.203
		9.6	10.1	0.052	0.157	0.209
Screw-drivers	Gas	25.7	27.6	0.288	0.430	0.718
	Coke	40.6	50.7	0.363	0.787	1.149
	Coal	45.1	67.4	0.493	1.052	1.545

The heating of the furnace took practically no time with gas, and from 40 to 110 min. with coke and coal respectively.

[1] On the basis of Pf. 56.2, or $0.128 per hour.

$$\frac{a(z_b - z_a)}{z'} \cdot \frac{z_c}{z_d}$$

To the rotation of the crank k through the angle a corresponds therefore a rotation of the spurwheel d through the angle

$$a\left[\left(\frac{z_d - z_c}{z_d}\right) + \left(\frac{z_b - z_a}{z_b}\right) \cdot \frac{z_c}{z_d}\right]$$

Example: let z_a be 80, $z_b = 100$, $z_c = 80$, and $z_d = 60$. Then

$$a\left[\frac{60-80}{60} + \left(\frac{100-80}{100}\right) \cdot \frac{80}{60}\right] = \left(-\frac{1}{3} + \frac{1}{5} \cdot \frac{4}{3}\right)a = -\frac{1}{15}a$$

Or the shaft rigidly connected with the wheel d moves with a speed equal to 1/15th that of the driving shaft in opposite direction.

If in the above formula the following values be substituted: $z_a = z_c = z_b - 4$, $z_d = z_b - 8$, then the transmission ratio would be as follows:

$$i = -\frac{16}{2z_b - 8z_b} \cdot a$$

which shows that with the same pitch of teeth in all spurwheels and the same differences of diameters, but various ratios of numbers of teeth, any desired ratio of transmission may be obtained. If the first pair of wheels is larger than the second, the transmitted motion is in a direction opposite to that of the driving shaft, and vice versa. The rotation of the wheel *d* rigidly connected with the driven shaft is due to the fact that the parallel sliding motion of the wheel *c* communicates to it a motion in a direction opposite to that of the driving shaft. These two opposite motions transmitted to the wheel *d* at the same time, produce a relative rotation of the wheel *d* in which only a part of the angular speed of the driving shaft is transmitted to that wheel, the magnitude of the angular speed transmitted being a function of the eccentricity and wheel ratio. The nearer the two opposite motions are to each other, the greater is the ratio of transmission of the gear, so that even a ratio $1:\infty$ may be obtained by making the two directions of motion equal to each other.

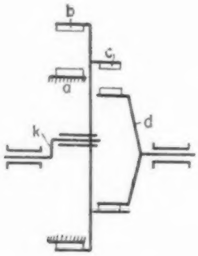


FIG. 5 KOMMER PLANETARY GEAR, SECOND TYPE

SCREW PILES (*Pieux à vis et vis d'encrage, Nouvelles annales de la construction*, 6th ser., vol. 9, p. 164, November 1912, 1 p., 19 figs. *d*). Description and drawings of various types of screw piles, as well as a short bibliography of same.

CONTRIBUTION TO THE THEORETIC STUDY OF FLEXURE (*Contribution à l'étude théorique de la flexion*, P. Keelhof. *Annales de l'Association des ingénieurs sortis des écoles spéciales de Gand*, ser. 5, vol. 5, no. 3, p. 274, 1912. 14 pp. *mt*). A mathematical investigation consisting in an application of the equation of Maurice Lévy to a determination of the cases where the usual formula of flexure gives correct results as far as stresses are concerned, whether shearing, or with respect to the moment of flexure; a short discussion is appended on elastic deformations.

THE LAW OF VISCOSITY OF FLUIDS (*Das Gesetz der Viskosität der Flüssigkeiten*, A. Batschinski. *Physikalische Zeits.*, vol. 13, no 24, p. 1157, December 1, 1912. ½ p., 1 fig. *et*). Advance notice of an article to be published. The author claims to have established that *the viscosity of a fluid is a function of its specific volume*. Attention is called to this notice because, as has been shown by Ubbelohde (*The Journal*, August 1912, p. 1247), " viscosity is the only factor affecting the coefficient of friction."

and all oils having the same viscosity have the same coefficient of friction in a bearing. It is often easier, however, to determine the specific volume of an oil than its viscosity, and the former may thus be used for the estimation of lubricating qualities of liquids.

SELECTION OF POWER (*Die Wahl einer Betriebskraft*, F. Barth. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 41 and 42, October 12 and 19, 1912, 15 pp., 1 fig. *cs*). The author does not attempt to establish any universal rules for the selection of power, but shows that for large power units, say over 6000 h.p. the steam turbine has to be used, while in other cases choice lies between a heat engine and an electric motor. Utilization of heat of exhaust gases in the case of internal-combustion engines is of advantage in the case of large gas engines in such as large steel works and mines, but one must not forget that when the engines work only intermittently, the supply of heat will also be intermittent. As the load factor of the plant and local fuel price decrease, the cheapest engine becomes the most economical, which means for large outputs at least, the steam engine. With a high load factor and high price of fuel the Diesel engine is to be preferred, but for power reserves and low load factor plants they are of advantage only in cases where particular stress is laid on the facility of, and speed in starting. Whether a suction gas plant is advisable in any particular case, is mainly a matter of the price of fuel: it is, when there is available fuel at a low price per heat unit, while in large plants the preliminary gasification of the fuel is of advantage in that it permits of the recovery of such valuable by-products as nitrogen and tar. The author shows that the introduction of the naphthalene, Diesel and suction gas engines, and steam superheat in steam engine plants have deprived the central station of a large amount of its previous importance, since the isolated plant is now about as economical in its operation as the large central plant, particularly since it does not have to meet the expense of laying the cable net. The author goes so far as to state that the central plant can compete in the cost of producing power with the Diesel engine only where it has a steady market for its waste heat.

The article contains a large amount of interesting statistical material.

Pumps and Fans

CALCULATION OF CENTRIFUGAL PUMPS (*Berekening van de centrifugaal-pomp*, P. F. A. von Wolzogen Kühr. *De Ingenieur*, vol. 27, no. 45, p. 897, November 9, 1912. 5½ pp., 6 figs. *tA*). An attempt to give an exact and practical theory of centrifugal fans without using Euler's fundamental formulae. Fig. 6 is a cross-section, normal to the axis of a fan. The polar equations of motion are:

$$dm\,(r''-r\theta'^2) = -dp.\,sb \qquad [1]$$

$$\frac{dm}{r}\,\frac{d(r^2\theta)}{dt} = dT \qquad [2]$$

where $w_u = r\theta$ is the absolute tangential velocity, and $w_r = r''$ absolute radial velocity. From the equation of continuity it follows that $sb.w_r =$ constant of $rb.w_r =$ constant, since s (length of the element) is proportional to r. Equation [2] can also be written as

$$rdT = \frac{dm}{dt} d\ (w_n r) \qquad [3]$$

and since $rdT = dM$, a moment, and $\frac{dm}{dt}$ is the mass flowing per second, which in the stationary state is constant and equal to M_{sec} [3] may be written as

$$dM = M_{sec}.d(w_n r) \qquad [3a]$$

which, on being integrated, gives

$$M = M_{sec}.\ [(w_n r)_2 - (w_n r)_1] \qquad [4]$$

From equation [1] the author derives

$$w_r dw_r - \frac{w_n^2}{r} dr + \frac{g}{\gamma} dp = 0 \qquad [5]$$

which permits of the calculation of the increase in pressure when w_n is known as a function of r. The tangential component w_n consists of $\omega\ r$

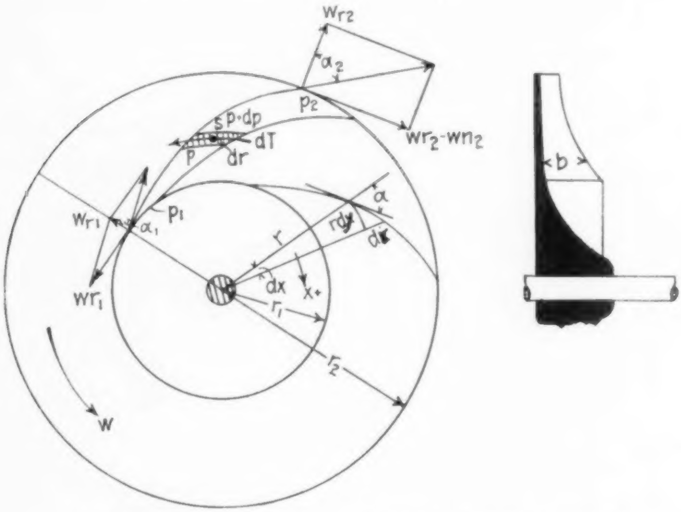


Fig. 6 Fan Wheel, Section of Blade and Stress Diagram

where ω is the angular velocity of the axis, and of relative velocity $r\chi$, the origin of this velocity component in the tangential direction being the deflection of its blade curvature from the radial direction, so that

$$w_n = (\omega + \chi') \quad \text{or} \quad w_n = (\omega - \chi') \qquad [6]$$

the latter for the case when χ is in the direction of rotation opposite to that of ω .

Further

$$r\frac{d\chi}{dt} = r\frac{d\chi}{dr} \cdot \frac{dr}{dt} = \omega r - w_n \quad \text{and} \quad r\frac{d\chi}{dr} = \frac{\omega r - w_n}{w_r} \qquad [8]$$

But $r\,\frac{d\chi}{dr} = tg\ \alpha$, and therefore

$$tg\ a = \frac{\omega r - w_n}{w_r} \qquad [8a]$$

And since w_n at entrance and exit are fixed, the following two equations hold:

$$tg\ a_1 = \frac{\omega r_1}{w_{r1}} \qquad [8b]$$

and

$$tg\ a_2 = \frac{\omega r_2 - w_{n2}}{w_{r2}} \qquad [8c]$$

The second of these equations may also be written in the form

$$tg\ a_2 = \frac{(\omega r_2)^2 - \omega(w_n r)_2}{\omega r_2\, w_{r2}} = \frac{u_2^2 - C}{u_2\, w_{r2}} \qquad [8d]$$

if H. Lorenz's value for the wheel constant $C = \frac{1}{\eta}\left(gh + \frac{c^2}{2}\right)$ be accepted; u_2 is here the peripheral speed of the fan.

The radii of curvature of the blades may be taken with reference to the angles φ_1, and φ_2, in which case w_n becomes known as a function of r, and vice versa.

From [8c] it follows that when r_2 is small and w_{n_2} large, then a_2 must be negative (leading blade); in the opposite case a_2 is positive (lagging blade). To be able to realize any desired angle, the simplest way is to select for the blade curvature an arc of a circle. It is further unnecessary to determine w_n as a function of r in order to determine the increase in pressure; the author indicates another method, but develops it only partly.

The author investigates further the various forms of fan blades, gives a graphical method for the construction of their radius of curvatures, as well as a discussion of the processes of flow of the fluid in the fan.

Steam Engineering

New Boiler Construction for Steam Turbine Plants with Special Regard to Water Tube Boilers. (*Neuere Dampfkessel Konstruktionen für Dampfturbinenkraftwerke unter besonderer Berücksichtigung der Steilrohrkessel*, Fr. Münzinger. *Zeits. für das gesamte Turbinenwesen* vol. 9, no. 31, p. 488, November 10, 1912. (Continuation of a series of articles). Description and drawings of the Borsig high capacity boiler and of the Glogner boiler. Although with a considerable expenditure of time and labor the headers of two-header tubulous boilers may be made so as to give entire satisfaction, it is a matter requiring care, and one that cannot be handled by the methods of mass production. Babcock & Wilcox have introduced the sectional construction which permits of rapidly making up a boiler of any height, but even this method is expensive on account of numerous tube joints. To avoid this, Engineer Glogner of Charlottenburg, Germany, uses instead of numerous tubes of small diameter a few tubes of comparatively large diameter carrying inside tubes of smaller diameter and having their ends beaded in horizontal sectional headers (Fig. 7A);

the front ends of the inner tubes are provided with reducing sleeves which are beaded in the front headers while the rear ends are held tight by nickel steel nuts (in the first Glogner boilers these nuts were protected by cast-iron caps, but experience has shown that such protection is superfluous). To clean the boiler, the nuts are loosened, and the inner tubes removed by a blow of a wooden mallet; in putting the boiler together again it is sufficient to have the inner tube loosely beaded in: the path of the gases is so arranged that the temperature of the external wider tube is higher than that of the inner tubes, and the former in expanding makes a tight joint. Fig. 7B is a section through the boiler; the return path of water is through tubes beaded in the upper part of the boiler, and there-

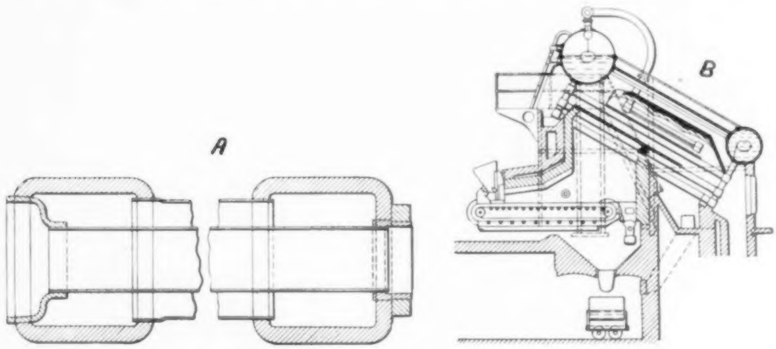


FIG. 7 GLOGNER WATER TUBE BOILER AND BOILER TUBE

fore heated but little. It is claimed that the Glogner boiler is of greater simplicity than the usual watertube boiler: where the latter with a heating surface of 250 qm (say 2500 sq. ft.) has to have 150 tubes with 300 joints, a Glogner boiler of equal capacity has only 48 tubes with 48 nuts. In testing a Glogner boiler in 1910, when this type was still to some extent in the experimental stage, Professor Josse found a boiler efficiency of 75 per cent, temperature of feedwater 50 deg. cent. (122 deg. fahr.), that of superheated steam 283 deg. cent. (541.4 deg. fahr.), and of flue gases 296 deg. cent. (564.8 deg. fahr.), the absolute steam pressure being 12.3 atmospheres.

HIGH CAPACITY WATER TUBE BOILER PLANT OF THE MUNICIPAL CENTRAL STATION IN BRANDENBURG A.H. (*Hochleistungs-Wasserrohrkessel-Anlage im Elektrizitätswerke der Stadt Brandenburg a.H.*, Mahr. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 42, p. 1708, October 19, 1912. 3 pp., 6 figs. *de*). Data from tests of 2 high capacity marine type boilers installed at the municipal plant of the city of Brandenburg, Germany. The construction of the boilers permitted the maximum production of 480 kg. steam per qm (9.9 lb. per sq. ft.) of ground area, while 1 sq. ft. of boiler heating surface required only 0.0725 sq. ft. of ground area. This shows the advantages of this type of boilers for use in places where the ground value forms a particularly important item in the cost of the station. The efficiency of the boiler plant is given as 79.0 per cent without feed heater, and 83.3 per cent with feed heater.

CORROSIONS IN STEAM BOILERS (*Des corrosions dans les chaudières à vapeur*, La Maestra. *La Technique moderne*, vol. 5, no. 10, p. 357, November 15, 1912, 6 pp. *ptA*). The author begins with the well-known fact that steam boilers are most affected by corrosion at the transverse riveting seams and at the curvature of the stamped parts; it is also very usual, particularly in the case of locomotive boilers, to find strongly corroded places on the inside of the boiler opposite its intermediate supports. Forney stated that when steel and iron are subject to corrosion and in parts submitted to powerful tensions at the same time, the corrosion will be stronger at the places of greater tension. From a study of a number of locomotives the author found that when the supports adhere closely when cold, and consequently produce a strong tension when the locomotive is hot, there are clearly pronounced corrosions along the supports, from 4 sometimes to as much as 7 mm (0.157 to 0.229 in.) deep, but when there was a play of about 1.50 mm (say 0.06 in.) between the boiler and its support when cold, no such corrosion was observed. The same explanation is given for corrosion affecting the stamped end-sheets of boilers. Stamping is at best a delicate operation, and the end sheets are subject to great tensions, so that if the work is not done well, or the material is somewhat defective, corrosion is likely to start.

On the same basis of corrosion being a function of the stress in the respective boiler sheet, the author proceeds to establish a formula for the speed of corrosion and time which it takes for corrosion to penetrate to a certain depth, e.g. that at which it becomes dangerous for the boiler. He derives the following two formulae:

$$V = P\frac{C}{s-y} \quad \text{and} \quad x = \frac{y}{2PC}(2s-y)$$

where V is the speed of corrosion, P pressure per unit, C constant $= \frac{PD}{2}$, s thickness of sheet, y depth of corrosion. It is worth noticing that the second equation, the law of the progress of corrosion, is that of a parabola.

To find how long it will take for corrosion to reach a certain depth, the author assumes that if in time t_1 a corrosion of depth h_1 is produced, to produce a corrosion of depth h_n, it will require some time $t_n = Kt_1$, where K is a certain coefficient to be determined. He further shows that K is the ratio of proportionality.

$$\frac{x_{h_n}}{x_{h_1}} = \frac{x'_{h_n}}{x'_{h_1}} = K$$

K has been determined experimentally for various values of the depth of corrosion h and thickness of plate s, and brought together in Table 4. The following example is quoted to show how this table and method are to be used. A locomotive boiler has cylindrical sheets 15 mm (0.59 in.) thick, with a diameter of 1315 mm (51.5 in.), rated at 10 kg (142 lb. per sq. in.); the boiler is subjected to a stress of 4.38 kg (62.2 lb. per sq. in.), while the maximum stress is 7 kg (99.4 lb. per sq. in.). The sheets may therefore be used until they have attained the minimum thickness of 9.39 mm.

but to simplify the calculation the depth of permissible maximum corrosion is found to have corroded to a depth of 3 mm (0.118 in.). By substituting is assumed here to be 6 mm (0.236 in.). After 4 years of work, the boiler values from Table 4 for K and the above equation for x, the following are found:

$$t_4 = \frac{104}{81} 0.48 = 61 \text{ (5 years and 1 month)}$$

$$t_5 = \frac{125}{81} 0.48 = 74 \text{ (6 years and 2 months)}$$

$$t_6 = \frac{144}{81} 0.48 = 85 \text{ (7 years and 1 month)}$$

Or after three years and one month more the corrosion in the boiler will pass the safe limit. In a supplementary discussion the author attempts

TABLE 4 TIME COEFFICIENTS FOR CORROSION IN BOILERS

Depth of Corrosion	Time Coefficient for Thickness of Plate s in Mm.					
	14	15	16	17	18	19
1	27	29	31	33	35	37
2	52	56	60	64	68	72
3	75	81	87	93	99	105
4	96	104	112	120	128	136
5	115	125	135	145	155	165
6	132	144	156	168	180	192
7	147	161	175	189	203	217
8	...	176	192	208	224	240
9	...	...	...	225	243	261
10	...	...	...	...	...	280

to trace the spread of corrosion rifts to original deficiencies in the plate due to irregularities in casting.

The author shows how to apply these equations to the determination of the most economic thickness of plates for a given pressure, or pressure for a given thickness of plates. A boiler has to be made to work with a pressure p, to be of diameter D, R being the coefficient of resistance of the plates, and the thickness has to be selected between s and s_1, the latter being larger; let t be the time required for corrosion to the depth h to be produced, and t_1 the time for corrosion to the depth h_1 such that $s-h=s_1-h_1$; then, as the author shows,

$$\frac{t_1}{t} - 1 = \frac{(s_1 - s + h)\,(s_1 + s - h)}{h(2s - h)} - 1$$

by which the advantage of a thicker or thinner plate may be determined. Sometimes it may be of advantage to adopt the thicker plate only for the two lower half rings of the cylindrical part of the boiler instead of copper rings which are used for protecting that part of the boiler from too rapid corrosion.

The above applies in the first place to locomotive boilers.

Thermodynamics

A Technical Process for the Determination of Heat Conduction in Plate-Shaped Bodies (*Ein technisches Verfahren zur Ermittlung der Wärmeleitfahigkeit plattenförmiger Stoffe*, R. Poensgen. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 41, p. 1653, October 12, 1912. 5 pp., 10 figs. *eA*). The Nusselt process for the determination of the coefficient of heat conductivity (the heating element is placed in the center of a ball which is filled with the material under test; the coefficient of heat conductivities is determined from the amount of heat produced at the heating element which is in turn determined from the strength of the heating current and electrical pressure, and from the amount of heat transmitted to the surface of the ball) can be used only for materials plastic enough to fill a ball, and having equal coefficients of heat transmission in all directions. For materials which have to be tested for heat transmission in the shape of plates Gröber used

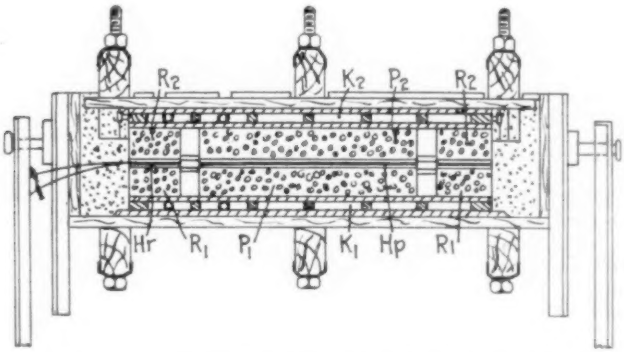


Fig. 8 Poensgen Arrangement for Determining the Coefficient of Heat Transmission of Plate Shaped Bodies

an arrangement in which two layers of the material under test enclosed a plate-shaped heating element; the disadvantage of this arrangement lay in the escape of heat sidewise along the edges of the heating element not covered by the plates of material under test. To avoid this the author encloses the main heating element into an auxiliary one having the same temperature as the first and arranged to prevent the sidewise escape of heat without materially affecting the amount transmitted through the plates of the material tested. In Fig. 8 H_p is a square heating element lying between two plates P_1 and P_2 of material under test, of equal thicknesses and of the same area as the heating element. Around the last and at a distance of 2.5 to 4 cm (0.97 to 1.57 in.) is placed a ring-shaped heating element H_r, 11 cm (4.3 in.) broad, covered on both sides by plates R_1 and R_2 of some heat insulating material of the same thickness as the plates under test. The other sides of the plates P and the rings R are covered by water cooled jackets K_1 and K_2. Care should be taken that the temperature of H_p should be equal to that of H_r, as well as that the temperatures of K_1 and K_2 be equal to one another. Under these

conditions the isotherms ought to lie in the planes parallel to the surface of the plate, and the coefficient of heat conductivity may be determined from the formula:

$$\lambda = \frac{Q \cdot \delta}{2F(t_1 - t_2)}$$

where Q is the amount of heat supplied to the heating element in calories, t_1 and t_2 are the temperatures of the warmer and colder surfaces of the plates under test in deg. cent., δ their thickness in m. and F their area in qm.

For heating elements the author used fabrics woven of constantan wire and asbestos fiber. The article contains data from tests made by the author

TABLE 5 COEFFICIENTS OF HEAT TRANSMISSION

Material	Specific Weight		Coefficient of Heat Transmission at 20 Deg. Cent.
	Kg/Cbm	Lb. per Cu. Ft.	
Linoleum	1183	74	0.16
Wood normal to the fiber	546	34.2	0.13
Wood parallel to the fiber	551	34.6	0.30
Teak wood normal to fiber	642	40.5	0.15
Teak wood parallel to fiber	604	37.8	0.32
Oak normal to fiber	825	52	0.18
Oak parallel to fiber	819	51.2	0.31
Asbestos slate	1783	112	0.19
Gypsum plates with enclosed pieces of cork	685	43	0.25
Building gypsum	1250	78.5	0.37
Wood and brick building wall	...	...	0.28
Machine made brick	1672	106	0.45
Brick wall	1850	116.2	0.35
Concrete (1:4, dry)	2180	136.5	0.65
Concrete (1:12, fresh)	2050	128	0.70

at the laboratory of the Technical High School in Munich, of which the most important are reproduced in Table 5 (cp. a similar table from the tests of Biquart, *The Journal*, December 1912, p. 2105).

Among other things established by the author was a confirmation of the fact found by Nusselt, viz. that the coefficient of heat transmission increases as the temperature rises.

Miscellanea

PROGRESS IN REFUSE DESTRUCTOR FURNACE ENGINEERING (*Fortschritte auf dem Gebiet der Müllverbrennung*, H. Schaefer. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 48, p. 505, November 29, 1912. 3 pp., 10 figs. *d*). The great difficulty in refuse destructor furnace work was the *method of handling ashes*, a problem all the more difficult owing to the fact that the ashes and dust formed from 6 to 10 per cent of the weight of the refuse burned. Handling by manual labor proved to be both an expensive and an unhygienic proceeding. A suction fan placed between

the ashpit and exit was experimented with, but the hot ash and dust handled was destructive to the fan blades. An attempt to meet the difficulty by placing the fan behind the exit proved to be equally unsuccessful owing to the trouble experienced in keeping the installation in working order. Hartmann & Co., of Offenbach a.M., Germany, have devised an improved system described in the present article, the main feature of which consists in producing the suction by a specially constructed pump (no details as to its construction are given), and in moving the dust and ashes through closed pipes exclusively. The article describes in some detail the piping system in the Hartmann installation in the municipal refuse destructor plant in Frankfort, where it was found to give entire satisfaction.

MECHANICAL TILLING AT THE BOURGES EXHIBITION (*Le labourage mécanique a l'Exposition de Bourges*, H. Pillaud, *La Technique moderne*,

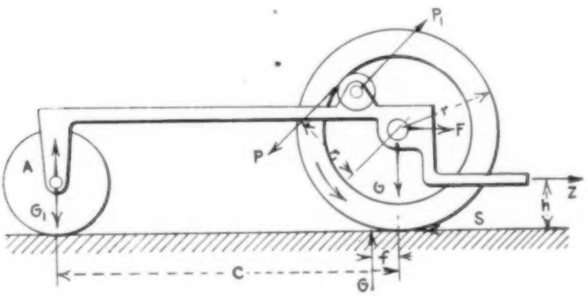


FIG. 9 DIAGRAM OF STRESSES IN A MOTOR PLOW

vol. 5, no. 9, p. 310, November 1, 1912. 6 pp., 5 figs. *adA.*). A very interesting account of the competitive tests of *farm tractors* at the Bourges Exhibition (France, October 1912), together with a description of the best types of French tractors. The tests were made under various conditions, closely approaching those of actual work (e.g. in muddy and rocky ground) and were on the whole very satisfactory. Both internal-combustion engined and steam tractors were represented.

THE MOTOR PLOW FROM THE POINT OF VIEW OF THE AUTOMOBILE ENGINEER (*Der Motorpflug vom Standpunkt des Automobiltechnikers*, K. S. *Auto-Technik*, vol. 1, no. 19, p. 35, October 11, 1912, appended to *Allgemeine Automobil-Technik*, vol. 4, no. 41 *dt*). Examination of the principles of design of motor plows, mainly with respect to their difference from that of motor cars, with discussion of the construction of the motor, cooling, transmission gear, etc. Of interest is the discussion of the difference in the action of the flywheel in the two cases, as illustrated by the following numerical example. Let an automobile weigh 1000 kg (2200 lb.), and move at 70 km (43.5 miles) per hour. Let the motor make 1800 r.p.m., the weight of the flywheel be 25 kg (55 lb.), and its diameter through the center of gravity 400 mm (15.7 in.). The kinetic energy of the whole car is then:

$$\frac{mv^2}{2}=\frac{1000}{2\times 9.81}\cdot\left(\frac{70}{3.6}\right)^2=19200 \text{ mkg.}$$

That of the flywheel:

$$\frac{m_1v_1^2}{2}=\frac{25}{2\times 9.81}\cdot\left(\frac{3.14\times 0.4\times 1800}{60}\right)^2=1810 \text{ mkg.}$$

The tractor weighs 10000 kg (22000 lb.), and has a speed of 1 m (3.28 ft.) per sec. Its motor makes 350 r.p.m., while the weight of the motor is 900 kg (1984 lb.), and its radius 1 m (3.28 ft.). The kinetic energy of the tractor is then:

$$\frac{mv^2}{2}=\frac{10000}{2\times 9.81}\cdot 1=510 \text{ mkg.}$$

while that of the flywheel is:

$$\frac{m_1v_1^2}{2}=\frac{900}{2\times 9.81}\cdot\left(\frac{3.14\times 1\times 350}{60}\right)=15400 \text{ mkg.}$$

This comparison shows that while in the case of the automobile the resistances are supposed to be taken care of by the kinetic energy of the whole car, and the kinetic energy of the flywheel supplies only about one-tenth of it, the kinetic energy of the flywheel in the tractor is about thirty times that of the tractor itself. That indicates also the kind of motor best adapted for tractor work; if the total energy of the above flywheel were given up in one second, that would be equivalent to an output of about 200 h.p., or nearly four times the regular motor output: this shows the advantage of using multicylinder motors which can be equipped with lighter flywheels, and still produce a uniform turning moment.

The author also calls attention to the necessity of carefully considering in the case of a motor plow the variations of the vertical axial pressure due to the driving and external forces. When the motion is uniform, in addition to the axial load G (Fig. 9) there are acting the crown pressure P and the tractive effort Z on the rear wheel. The rolling resistance of the front wheels produces the force F acting backwards, the moment of the resistance at road surface of the rear wheels being equal to $G.f$. On the wheel circumference is acting the adhesion force S. The horizontal force acting on the axis is therefore $F+Z=S$, and the driving moment is:

$$P\,.\,r_1=G\,.\,f+S\,.\,r$$

The pressure on the teeth P of the small driving toothed wheel produces an equal but oppositely directed reaction P_1 on the bearing.

The tractive effort Z, when applied as is usually done, above the point of support of the carriage, produces a decrease of load on the front axle, by producing a moment $Z.h$ tending to lift the front part of the tractor off the ground.

The following numerical example may show with forces of what magnitude one has to deal in the case of tractors. A tractor of 9000 kg (say 20000 lb.) exerts, when at rest, a pressure on the road surface in front equal to 3000 kg (say 6600 lb.) and behind 6000 (say 13400 lb.). The axle base is 4 m (13.1 ft.), the diameter of the driving wheels 2 m (6.56 ft.). The motor capacity is 50 h.p., the efficiency of the plant down to the driving wheels is 0.8, so that 40 h.p. are transmitted to the rear axle. The speed

is 1 m (3.28 ft.) per sec. Of the motor output 50 per cent are consumed by the resistances of the tractor itself, 50 per cent are used to drive the plow.

Schiess Automatic Fire Alarm and Extinguisher (*Extincteur-avertisseur automatique d'incendie, système Schiess. Le Génie Civil*, vol. 61, no. 23, p. 467. October 5, 1912. ½ p., 1 fig. *d*). The apparatus is shown in Fig. 10. When the temperature rises beyond a certain point, the ther-

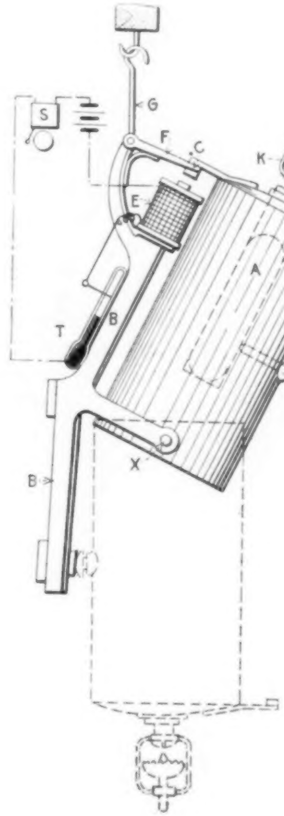


Fig. 10 Schiess Automatic Fire Alarm and Extinguisher

mometer *T* closes the current, and makes the electric bell *S* ring, and the electromagnet *E* energized. The electromagnet attracts the soft iron piece *F* which releases the catch hook *C*, and this lets the reservoir *R* tilt down about the axis *X*. During this tilting of *R* the percussor *P* breaks the glass tube *A* filled with hydrochloric acid which flows into the reservoir *R* filled with carbonate of soda, and produces an energetic evolution of carbon dioxide. This gas driven by the pressure inside the cylinder flows

with considerable violence through the opening *I*, and striking the shield *K*, spreads brushwise to all sides. It would appear, however, that some of the mixture of the acid and carbonate of soda would be carried away by the gas and thrown to all sides, which might be undesirable under certain conditions.

On the Installation and Operation of Wood-Drying Plants (*Uber die Einrichtung und den Betrieb von Holztrocknungsanlagen*, Max Dribbusch. *Werkstattstechnik*, vol. 6, no. 19, p. 497. October 1, 1912. 5 pp., 14 figs. *p*). A discussion of the principles of installation and operation of wood-drying plants written in a thoroughly practical manner; particular requirements for the more important kinds of wood are indicated, and the advantages and disadvantages of various systems of compressed air application discussed.

Safety Appliances for Use with Tanks for the Storage of Explosive Liquids: Automatic Extinguishing Devices, Dr. Flach's System, patented in Germany (*Sicherheitsmassnahmen bei Tanks für Lagerung feuergefährlicher Flüssigkeiten durch selbsttätig wirkende Löschvorrichtungen System Dr. Flachs D.R.P.*, E. Schultze. *Allgemeine österreichische Chemiker- u. Techniker-Zeitung*, vol. 30, no. 20, p. 153. October 15, 1912. 2½ pp., 1 fig. *d*). Description of a new system of *fireproof tanks* for the storage of dangerous liquids like benzine or gasolene, based on a combination of the principle of the Davy safety lamp with that of producing low temperatures by expanding compressed non-oxidizing protective gases. The surface of the liquid is protected by one or more fine wire screens which take care of temperature variations, prevent the explosions of the liquid, and act as cooling surfaces. In addition, the pressure produced by the small explosions under the wire screen (just as in a Davy lamp) automatically sets the tank in connection with steel flasks filled with non-oxidizing gases (carbon dioxide, nitrogen); these gases expand on reaching the wire net, and thereby produce a powerful cooling which prevents the spread of fire, and cools off the wire net itself. This system has been approved by the Berlin factory inspectors.

Submarine Sled for Dräger Divers (*Unterseeschlitten für Dräger-Taucher. Periodische Mitteilungen aus dem Draegerwerk Lübeck*, no. 4, October 1912. *d*). The construction of diving apparatus without air pipes, such as the apparatus of the Drägerwerk Lubeck (The Journal, October 1912, p. 1584) and Westfalia (The Journal, November 1912, p. 1900) made the diver to a considerable extent independent of the boat acting as his base of operation, and this made it desirable to devise some means for making his locomotion more rapid than walking along the bottom, which would give him more time under water for placing torpedoes, inspecting the location of submarine obstructions to navigation, searching for wrecks, etc. The Drägerwerk Lubeck designed for this purpose what they call a *submarine sled*, or a boat provided with air tanks and vertical rudder, both arranged in such a way that by suitably operating them the boat may be either kept at the surface, or lowered to the bottom. Cp. *Scientific American*, December 21, 1912, p. 534.

The Education of Molders and Founders at the Royal Trade School

FOR THE METAL INDUSTRY AT ISERLOHN (*Die Ausbildung der Former und Metalgiesser an der Königl. Fachschule für Metalindustrie zu Iserlohn*, H. Krause. *Giesserei-Zeitung*, vol. 9 no. 20, p. 623. October 15, 1912. 6 pp., 5 figs. *d*). Description of the course for molders and founders at the Iserlohn Trade School. The course in the school is equivalent to a period of apprenticeship, and therefore the program includes a large amount of practical work, and avoids all theory that is not of recognized value to the practical foundryman. The program of the school is quoted in detail.

DURALUMIN (*Duralumin*, L. M. Cohn. *Elektrotechnik und Maschinenbau*, vol. 30, nos. 39 and 40, pp. 809 and 829, September 29 and October 6, 1912. 10 pp., 11 figs. *ep*). Data from an investigation of the properties of *duralumin* previously published by the same author in *Zeits. des Vereines zur Beförderung des Gewerbefleisses*, 1911, no. 1. The first part of the article (in no. 39) contains among other things information as to substances attacking duralumin; the second part (in no. 40) is devoted to its use in electric conductors.

Supplementary References

EXPERIMENTS OF THE FIRM A. BORSIG IN TEGEL NEAR BERLIN WITH DUPLEX-SPRING VALVES IN HIGH-SPEED WATER PUMPS (*The Journal*, October 1912, p. 1577). A long explanation why some pumps with both valves positively loaded do start, and why they did not in the tests made by the author: *Die Fördertechnik*, vol. 5, no. 11, p. 262, November 1912.

NEW EXPERIMENTS ON THE COMBUSTION OF NITROGEN IN EXPLODING GAS MIXTURES (*The Journal*, October 1912, p. 1564). Description of the application of this process to the production of nitric acid from atmospheric nitrogen, and apparatus used, in *Industria*, vol. 26, no. 46, p. 739, November 17, 1912.

VALVELESS PUMP "BAVARIA" (*The Journal*, August 1912, p. 1253). Detailed description and drawings in *Pompe sans soupape*, *Portefeuille economique des machines*, 5th ser., vol. 10, November 1912, p. 170.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XXIII)

ARTICLES IN PERIODICALS[1]

GASOLINE CAR-FERRY FOR AN ELECTRIC RAILWAY. *Engineering News*, November 21, 1912. 3 pp., 7 figs.

Car-ferry and transfer boat used on the Ohio River between Evansville, Ind. and Henderson, Ky.

GAS PRESSURE REGULATIONS OF THE PUBLIC SERVICE COMMISSION FOR COMPANIES IN BOROUGH MANHATTAN, NEW YORK CITY. *Engineering News*, October 31, 1912. 2½ pp.

LIQUID FUELS; THEIR PRODUCTION PECULIARITIES AND INVESTIGATION, L. Schmitz. *Zeitschrift des Vereines deutscher Ingenieure*, October 26, 1912. ⅙ p.

A review of a book of the above title.

MOTOR SHIP EAVESTON, THE SINGLE SCREW. *The Engineer* (London), October 25, 1912. 5 pp., 9 figs., 1 curve. *dpA*.

This ship is 275 ft. 9 in. long, 40 ft. 6 in. beam, 4310 tons displacement, fitted with an 800 h.p. Diesel engine 90 r.p.m. at 9 knots.

MOTOR SHIP ROLANDSECK. *The Engineer* (London), November 22, 1912. 2½ pp., 6 figs. *dpA*.

Note and description of the Carels-Tecklenborg Diesel engines while on trial trip.

OIL AS FUEL, THE USE OF, A. O. Krieger. *Engineering News*, November 21, 1912. 1 p.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.

REPORTS OF MEETINGS

BOSTON MEETING, DECEMBER 20

A meeting of the Society was held at Boston, Mass., on December 20, when Prof. H. W. Hayward of Massachusetts Institute of Technology presented a paper on The Testing Laboratory and the Constructing Engineer. He described the equipment and functions of the different types of testing laboratories (works, government, technical schools, etc.), and brought out various misconceptions prevailing even among intelligent engineers as to what can and cannot be done in the laboratory, and as to what constitutes a reasonable specification for any particular class of material or structure. The value of the laboratory in determining compliance with specifications, also the value of laboratory tests in furnishing practical standards for incorporation in specifications, were indicated at some length, and a list of problems was given which had actually been put up to laboratories to solve. Examples were given of the amusing and absurd requirements which occasionally creep into well regulated specifications, and a number of slides were shown illustrating various points covered by the address. In closing the author appealed for a closer coöperation between the manufacturers and contractors and the laboratory to the end that more tests be made of full size specimens and of structures under conditions of actual use.

The paper was discussed at length by Prof. I. N. Hollis, Mem.Am.Soc. M.E., Prof. Geo. F. Swain, Mem.Am.Soc.M.E., and Mr. Woods.

STUDENT BRANCHES

ARMOUR INSTITUTE OF TECHNOLOGY

On December 4 the Armour Institute of Technology Student Branch held its third regular meeting. The speaker of the evening was Mr. S. Rosenzweig of the Erie City Iron Works, who presented an illustrated lecture on Poppet Valve Engines and Superheated Steam. Since the use of superheated steam is not general in this country the paper was of great interest to all and was discussed by Professors G. F. Gebhardt, Mem.Am.Soc.M.E., and H. L. Nachman, and A. W. Semerak.

COLUMBIA UNIVERSITY

The student branch at Columbia University held a meeting December 6, and Mr. Place of Warren and Whetmore, architects for the New York Central Railroad, gave a talk on the Mechanical Features of the New Grand Central Terminal.

CORNELL UNIVERSITY

Sibley College Student Branch held a meeting December 11 at which Prof. R. C. Carpenter, Mem.Am.Soc.M.E., spoke on The Practical Problems of Smoke and Dust Prevention.

LEHIGH UNIVERSITY

A meeting of the Lehigh University Student Branch was held November 19 when Professor Schweinitz was elected honorary chairman. The Electric Lighting of Steam Trains was presented by Mr. Johnson; Some Efficiency Secrets by Mr. Brooke; and Mr. Butterfield spoke briefly of the cleaning of blast furnace gas for use in engines; dry, wet and mechanical scrubbers were also discussed.

On December 10 Mr. Janeway presented a paper on An Economical Method for Producing Sound Steel, and Lieut. W. D. Greetham, government inspector for the navy at the Bethlehem Steel Works, spoke on Naval Ordinance and Gunnery.

POLYTECHNIC INSTITUTE OF BROOKLYN

At a meeting of the Polytechnic Institute Student Branch on December 7, Geo. A. Orrok, Mem.Am.Soc.M.E., lectured on Hydro-Electric Development with particular reference to the water power plant being built near Chattanooga on the Tennessee River. Discussion was offered by Professor Ennis, Mem.Am.Soc.M.E., Messrs. Adler, Heustis and other student members.

PURDUE UNIVERSITY

An unusual interest is being taken in the Purdue University Student Branch this year. Regular meetings are held every two weeks and are well attended. The meeting on December 3 was addressed by Mr. Shaw, western sales manager of the Westinghouse Company, who gave the history of turbine development, discussed features of the construction of the most modern turbines and condensors, and gave a clear idea as to the field of the turbine as a prime mover.

STEVENS INSTITUTE OF TECHNOLOGY

Stevens Institute of Technology Student Branch held a meeting November 26 at which Prof. F. R. Hutton, Honorary Secretary of the Society, read a paper on The Testing of the Automobile Motor. Secretary Calvin W. Rice told of the progress of the work of the student branches and Past-President Alex. C. Humphreys closed the meeting with a few appropriate remarks.

On December 6 a paper by Carl F. Dietz, Mem.Am.Soc.M.E., on the Manufacture of Abrasive Materials was presented. Messrs. Kinsey, Vander Veer, Smith, Aronson and Kepke entered into the discussion.

George R. Henderson, Mem.Am.Soc.M.E., read his paper on The Development of the Locomotive at the December 10 meeting. Messrs. Vander Veer and Henry took part in the discussion.

SYRACUSE UNIVERSITY

A student branch was established at Syracuse University December 3, 1912, and the following officers have been elected: Honorary Chairman, Prof. W. E. Ninde; Chairman, O. W. Sanderson; Corresponding Secretary, R. A. Sherwood.

UNIVERSITY OF ARKANSAS

At the December 2d meeting of the University of Arkansas Student Branch, Prof. B. C. Carnahan read a paper on Gas Producers, and Producer Gas Analysis, which was followed by a second by John Danner on The Setting of Steam Engines.

UNIVERSITY OF CINCINNATI

On December 3 Mr. Rosenzweig of the Erie City Iron Works read his paper on Superheated Steam and Poppet Valve Engines before the University of Cincinnati Student Branch. He showed the economy of superheated steam and the adaptability of the poppet valve to its use. His sixty lantern slides showed the detailed construction of engine and valve gear in particular. Messrs. A. K. DeLeeuw, J. B. Stanwood, B. S. Hughes, R. S. Brown and Prof. J. F. Faig, all members of the Society, were present and took part in the discussion.

UNIVERSITY OF KANSAS

The University of Kansas Student Branch held a meeting December 5 and several members of the senior class gave reports of the annual inspection trip to Keokuk, Iowa, where they visited the Mississippi River Power Company's hydro-electric plant. Mr. Knerr read a paper on The Economics of the Mississippi River Power Project at Keokuk, Iowa; Mr. Plank reported on the stream flow of the Mississippi River at this point, and Mr. Tangeman gave an account of the electrical installation and power distribution of the plant.

UNIVERSITY OF ILLINOIS

On December 6 the University of Illinois Student Branch held its regular bi-monthly meeting. Mr. S. Rosenzweig of the Erie City Iron Works read his paper on Poppet Valve Engines and Superheated Steam, following which there was an interesting and general discussion.

The members of The American Society of Mechanical Engineers at the University of Illinois, with the heads of the several engineering departments, gave a dinner to Dean W. F. M. Goss of the College of Engineering on December 13 at the University Club in Urbana, as an expression of their appreciation of the honor paid him by his election to the presidency of the Society. Among the guests present were the members of the Council of Administration of the University of Illinois and Dean Charles H. Benjamin of Purdue University. Prof. Ira O. Baker acted as toastmaster and the following toasts were given: The National Engineering Societies, Professor Arthur N. Talbot; Congratulations from the University, Dean Eugene Davenport of the College of Agriculture; Greetings from Purdue University, Dean Charles H. Benjamin; Response, Dean W. F. M. Goss.

On December 18, a special engineering convocation was held to permit the general faculty and the students in the College of Engineering to express their appreciation of the honor paid Dean Goss. Brief addresses were made by President Edmund J. James and Prof. A. N. Talbot, after which Prof. Ira O. Baker presented to Dean Goss a beautifully engrossed testimonial signed by representatives of the faculty and the various engineering organizations at the University of Illinois.

UNIVERSITY OF MISSOURI

The University of Missouri Student Branch held a meeting November 18 at which Messrs. Heileman and Klein read a paper on The Diesel Engine. On December 2 Messrs. Pierce and James presented a paper on The Present Day Aspect of Industrial Management. Short talks on the subject by Prof. Hibbard and Messrs. Wesson and Fessenden followed.

UNIVERSITY OF NEBRASKA

A meeting of the Student Branch of the University of Nebraska was held December 10 at which Messrs. Toney and Goddard presented The Purchase of Coal by Specification for discussion. The subject was treated under the following heads: origin of the purchase of coal by specification; disadvantages; advantages; methods of buying with data. Professor Baker criticised the paper.

UNIVERSITY OF WISCONSIN

Before a meeting of the University of Wisconsin Student Branch on December 12, Prof. C. C. Thomas gave an account of the New York meeting of the Society held November 12, when E. B. Passano's paper on Measuring Efficiency in Manufacturing was read.

WASHINGTON UNIVERSITY

A joint session of the Washington University student branches of the American Institute of Eelectrical Engineers and of the Society was held on December 11. Fred. H. Kohlmeyer of the mechanical engineering department read a paper on Scientific Shop Management, and C. E. Wright of the electrical engineering department gave a talk on the Transmission and Distribution System of the Illinois Traction System. A general discussion followed.

YALE UNIVERSITY

At a meeting of the Yale Mechanical Engineers Club held December 10, William Kent, Mem.Am.Soc.M.E., spoke on Engineering Opportunities.

NECROLOGY

RALPH P. BADEAU

Ralph P. Badeau was born in Brooklyn, N. Y., October 26, 1886, and received his early education at Rutgers College Preparatory School of New Brunswick, N. J., and at the Battin High School of Elizabeth. He was graduated from Stevens Institute of Technology in the class of 1909, and immediately afterward entered the employment of Purdy and Henderson, structural engineers of New York. While with them he had charge of the design of the McNeill Building, Chicago, Ill., and of the crane runway for Inland Steel Company. In November 1910 he became assistant superintendent of the A. & F. Brown Company of Elizabeth, N. J. where he was employed until his death, July 20, 1912.

GEORGE H. SCHULTE

George H. Schulte was born in Cincinnati, Ohio, April 16, 1848, and at the age of sixteen went to Little Rock, Ark., where he was employed in the Fones Brothers' hardware store. He became its manager in 1870. Twelve years later he went to Milwaukee, Wis. where for the next twenty years he was superintendent of the Milwaukee Harvester Company, having full charge of the mechanical part of the works. Under his management the harvester was brought to so high a degree of perfection that it found a world-wide market. The company was purchased by the International Harvester Company of America in 1904, and Mr. Schulte remained as vice-president and general manager for one year, when he was compelled to retire on account of ill health. From 1906 to 1908 he was director and general manager of the J. I. Case Threshing Machine Company of Racine, Wis. The last three years of his life were spent in retirement and traveling. He died July 29, 1912.

FREDERICK H. CRABTREE

Frederick H. Crabtree was born at Ebbw Vale, England, Oc-

tober 5, 1863, and died in Anaconda, Montana, August 3, 1912. His early education was obtained in the Cardiff High School, and in 1876 he entered as an apprentice the Blaina Iron Works of Blaina, England, where he remained until 1881. Subsequently he was employed by the Thames Iron & Shipbuilding Company, London; Crossley's Woolen Factory, Halifax; Blaina Blast Furnace Works; Ebbw Vale Steel Works; and as chief engineer in charge of the Pyle Works coal washing and coking plant, Blaina.

Mr. Crabtree came to America in 1888 and accepted a position as master mechanic with the Rock Springs Coal Company of the Union Pacific Railroad. In 1890 he moved to Anaconda, Mont., where he held a number of responsible positions, among which was the supervision of the installation of the machinery in a 50-ton gold mill for the Republic Reduction Company, Republic, Wash. The last position he filled was as assistant superintendent of power plants of the Washoe Smelter Company, from which he resigned three years ago.

Mr. Crabtree is the patentee of a number of devices. In 1887 he was granted an English patent on improvements on a rotary steam engine; in 1904 on a pneumatic massage apparatus; in 1905 on an automatic cut-off steam valve; in 1910 on a paper fastener and on an auxiliary rim and tire protector.

ARTHUR C. SCOTT

Arthur C. Scott was born January 27, 1870 at Northbridge, Mass. He moved to Worcester when eleven years old and obtained his education in the public high school there and the Worcester Polytechnic Institute. His first employment was in the freight office of the New York, Providence & Boston Railroad and two years later he went to work for the Draper Machine Tool Company of Worcester where he remained for seven years. In 1903 Mr. Scott entered the employ of the Norton Company as traveling salesman, and in 1909 was sent to Great Britain to sell grinding wheels and to investigate conditions abroad. He died November 22, 1912.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on the current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The list of "men available" is made up from members of the Society and good men not members. Information will be sent upon application.

POSITIONS AVAILABLE

01 Engineer with practical experience to supervise the running and construction of a large plant, as assistant to chief engineer and to represent him during his absence; must be able to take charge of drawing-room and design of buildings and machinery. Knowledge of manufacturing sugar preferred. State previous experience, age and salary desired. Apply through the Society.

02 First class steam engineer for a large steel plant in Pittsburgh district. Give experience in detail, education, age and salary wanted.

03 Young mechanical engineer of ability and experienced in the efficiency methods of shop practice and setting of rates, having a suitable personality for prosecuting this work, to act as assistant to superintendent in charge of work. Salary $100 to $150. Position one of promise.

04 Young man on time-study work for New Jersey concern; must be practical mechanic technically inclined, preferably one with experience in armature and coil winding and general electrical work.

05 Man for production department, experience in technical lines, preferably in electrical work. Salary $100 upward. Location, New Jersey.

06 Works manager, trained manufacturing executive, experienced in problems of management of men, efficiency and factory accounting, with college preparation in chemistry, and with experience in the management of chemical, electro-chemical or metallurgical industries.

07 Expert cost-reduction man, thoroughly experienced in modern shop practice, to devote entire time to development and application of cost reduction methods in Canadian plant, manufacturing air compressors, rock drills, hammer drills and minor allied lines. Apply through the Society.

08 Young engineer to handle technical work in the drawing-room and testing plant of Ohio concern; to be in charge of experimental work with opportunity to branch into the commercial side.

09 Mechanical engineer, technical graduate for perfection of invention

for railroad use. Experience in automatic signal devices preferred. Location, New York. Apply through the Society.

010 Technical man to handle the engineering positions, correspondence and office routine for New York concern engaged in employment work.

011 Draftsman specializing in photographic machinery and apparatus. New York concern.

012 Salesman for New York concern manufacturing instruments for indicating, recording and controlling temperature and pressure.

13 Southern university desires instructor in mechanical engineering to take position at once. Desires graduate of high-grade engineering college, who can teach machine design, mechanical laboratory, engines and boilers. To the right man assurance of steady advancement in salary and rank. Salary to start $1000.

014 Water meter expert and designer. Location New York. Apply through the Society.

015 Air compressor expert and designer. Location New York. Apply through the Society.

016 Efficiency engineer with full knowledge of machines, machine tools and the various and proper rates of cutting the different metals usually encountered in machine shop practice; his duties will be to deal entirely with the machine shop end of the plant. Apply through the Society.

017 Draftsman, accurate, rapid; must have experience on steel stamping, drawing and forming tools, other experience valuable. State experience, references, and salary desired.

MEN AVAILABLE

1 Graduate, mechanical engineer, ten years' experience in design, testing and operation of mechanical and electrical plant equipment, especially qualified to handle men, desires connection with manufacturing or power production company preferably in New York. At present employed in responsible executive position.

2 Technical graduate, age 37, practical mechanic with ten years' experience in executive capacity, mill engineering, power generation and transmission, etc. Desires position with large progressive concern in New England as factory engineer or works manager.

3 Position as manager or superintendent by Member. Thoroughly posted on modern factory methods and can show results in cost reduction; good executive. Prefers location north of Ohio River. At present employed.

4 Production manager familiar with machine tools, engines, automobiles, valves and fittings. Qualified to construct and apply special tools and machinery to assist in production. Knowledge of labor-saving management including time studies, planning, routing, shop costs and premium systems.

5 Graduate, mechanical engineer, fifteen years' experience, desires position as works manager or superintendent. Familiar with modern methods

of handling men and machines to produce profitable results. At present employed, but desires a position with greater opportunities. Can furnish good references.

6 Junior, age 30, technical graduate, with experience in design, construction, operation and maintenance of power plants and sub-stations, testing electrical and other machinery, desires responsible position with large manufacturing plant or contractors on power or hydroelectric plant work. Salary $175 to $200 per month.

7 Sales manager, desires to connect with concern where opportunity will be given to produce maximum results. Broad gage man with initiative and exceptional ability acquired by wide experience in mechanical and electrical manufacturing, having worked from drafting-room to superintendent in bronze, iron and steel foundry, and machine shop.

8 Sales engineer, with office and shop experience in designing, wishes to change. Experienced in plant lay-out and equipment in detail; graduate Munich Polytechnic. Will consider salary and commission basis. Desires New York location.

9 Position desired as works manager, preferably with new company; long experience as factory man and engineer; now head of engineering department of large company.

10 Position desired as mechanical and electrical superintendent to take charge of entire power department of manufacturing concern. Thoroughly understands practical, economic and reliable operation; ten years' experience.

11 Technical graduate, twenty-five years' experience in invention and design of power machinery especially gas and steam, desires position.

12 Mechanical engineer, technical graduate, five years' experience in works engineering, gas engines and pumps, desires position as assistant to executive or manager.

13 Technical graduate eighteen years' experience in shop, drafting-room and executive positions, desires position as chief engineer, resourceful in designing and working out new ideas. Salary $2500.

14 Junior, age 27, technical graduate, six years' experience erecting, operating, and testing large gas engines, and various types of producers; gas engine and producer design in connection with trouble and experimental work, desires position where such experience will be of value. Location immaterial.

15 Technical graduate, thirty years' experience in designing, building and managing mills for cotton, cotton oil and by-products. Competent in development of new schemes connected with the cotton industry from the plantation to the most remote by-product.

16 Junior member, age 29, married, five years' experience in machine shop and general manufacturing, three years' as works engineer, desires position as assistant superintendent or with firm doing power plant or hydroelectric work.

17 Engineer, age 39, technical education, executive ability, experienced in designing, erecting and operating large gas and oil engines and producers, also manufacturing steam engines and turbines; electrical experience.

18 Member, technical graduate, desires position as sales engineer, experienced as skilled mechanic, chief draftsman and superintendent of construction of power plant equipment.

19 Mechanical engineer or chief draftsman with executive ability. Thorough experience in steam power plant design, heating and ventilating, mechanical equipment of round-houses, etc. Technical graduate.

20 Mechanical engineer, Stevens Institute graduate. Experience in plant superintendence and construction, power house engineering, boiler design, manufacturing and sales. Acquainted with territory in vicinity of New York City.

21 Mechanical engineer, Cornell graduate, experience in construction and superintendence of manufacturing mills; familiar with construction and operation of plants for manufacture of explosives, also manufacture of nitric and sulphuric acids and nitrate of ammonia. Desires position in operating or engineering end of this line of work.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

ACCOUNT OF THE HISTORY AND PRESENT STATE OF GALVANISM, John Bostock. *London, 1818.* Gift of L. B. Lent.

AMERICAN MACHINIST GRINDING BOOK, F. H. Colvin and F. A. Stanley. *New York, McGraw-Hill Book Co., 1912.*

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Trans. 1880, vol. 1; 1881, vol. 2; 1882, vol. 3. *New York, 1892.*

AMERIKANISCHE HÜTTENWERKE, Peter Eyermann. (Sonderabdruck aus der "Zeit. des Oesterr. Ingenieur-und Architekten Vereines" 1912, Nr. 33.) *Wien, 1912.* Gift of author.

ARTISTIC BRIDGE DESIGN, H. G. Tyrrell. *Chicago, 1912.*

ATLANTIC DEEPER WATERWAYS ASSOCIATION. Resolutions adopted by and address on The New Jersey Link, F. W. Donnelly. *New London, Conn., 1912.* Gift of New Jersey Ship Canal Commission.

BAU UND BETRIEB VON PRALL-LUFTSCHIFFEN, Richard Basenach. *Frankfurt a. M., 1912.*

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